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A Review of the Options Available to the Forest Industry for Producing Electricity from Wood Residues

by

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TABLE OF CONTENTS

LIST OF TABLES v

LIST OF FIGURES ix

NOMENCLATURE AND DEFINITIONS xiii

SUMMARY xvi

I. INTRODUCTION 1

- A. Background 1
- B. Scope 1
- C. Format 2

II. POTENTIAL FOR ELECTRICITY GENERATION FROM WOOD RESIDUES 3

- A. Energy Requirements of Mills 3
  1. Sawmills 3
  2. Pulp and Paper Mills 5
  3. Plywood and Veneer Mills 5
  4. Specialty Products Mills 5
- B. Meeting Electricity Demands With Mill Residues 8
  1. Sawmills 10
  2. Pulp and Paper Mills 10
  3. Plywood and Veneer Mills 15
  4. Specialty Products Mills 15
  5. Thermal and Electrical Energy Generation From Residues 22
- C. Competing Sources of Electricity 27

III. ALTERNATIVES FOR GENERATION OF ELECTRICITY USING WOOD 35

- A. Concepts for Energy and Material Flows in Mills 35
- B. Systems for Electricity Generation 35
  1. System Options 35
  2. Steam Boilers 40
  3. Gasifiers and Pyrolyzers 48

4. Combustors (non-pressurized) 54
5. Pressurized Combustors 61
6. Pulverizers 63
7. Liquefaction Units 63
8. Bottoming Heat Recovery Boilers 64

C. Rationale for Selecting an Electricity Generating System 66
1. System Performance Characteristics 66
2. Operational Considerations 72

IV. WOOD CONVERSION UNITS 78

A. General 78

B. Types and Definitions of Wood Conversion Units 78

C. Combustors 79
1. Suspension Burners 81
2. Single Chamber Combustors 84
3. Multiple Chamber Combustors 88
4. Fluid-Bed Combustors 90

D. Gasifiers 96
1. Updraft Gasifiers 101
2. Downdraft Gasifiers 110
3. Stirred Bed Gasifiers 114
4. Fluid-Bed Gasifiers 122

E. Liquefaction Units 126

F. Pyrolyzers 130

G. Steam Boilers 130

V. POWER FLUID HEAT EXCHANGE UNITS 138

A. Heat Recovery Boilers 138
1. Steam Boilers 138
2. Organic Fluid Boilers 140

B. Hot Gas/Hot Oil Heat Exchangers 142

C. Hot Gas/Hot Air Heat Exchangers 143
1. Applications 143
2. Materials Selection 145
3. Manufacturers 146

VI. PRIME MOVERS FOR ELECTRICITY GENERATION 148

- A. Steam and Organic Vapour Turbines 148
  - 1. Cycle Configurations 148
  - 2. Performance Characteristics 148
  - 3. Manufacturers 160

- B. Gas Turbines 164
  - 1. Cycle Configurations 164
  - 2. Performance Characteristics 165
  - 3. Manufacturers 173

- C. Internal Combustion Engines 180
  - 1. Cycle Configurations and Engine Types 180
  - 2. Performance Characteristics 181
  - 3. Manufacturers 189

- D. Stirling Engines 193
  - 1. Cycle Configurations 193
  - 2. Operating Characteristics 195
  - 3. Manufacturers 197

- E. Steam Engines 202
  - 1. Cycle Configurations 202
  - 2. Operating Characteristics 204
  - 3. Manufacturers 208

- F. Advanced Direct Conversion Systems 211
  - 1. Fuel Cells 211
  - 2. Thermionic Converters 214

VII. ELECTRICITY GENERATING SYSTEMS SUPPLIERS 216

- A. Steam Boiler Systems 216
  - 1. General 216
  - 2. Costs 216
  - 3. Manufacturers 218

- B. Gasifier and Pyrolyzer Systems 220
  - 1. General 220
  - 2. Costs 221
  - 3. Manufacturers and Developers 222

- C. Non-Pressurized Combustor Systems 222
  - 1. General 222
  - 2. Manufacturers 226

D. Pressurized Combustor Systems 226
  1. Combustion Power Co. 226
  2. Aerospace Research Corp. 227

E. Bottoming Heat Recovery Boiler Systems 227

VIII. STATUS OF SYSTEMS FOR ELECTRICITY GENERATION 230

A. Commercial Systems 230

B. Emerging Systems 230

REFERENCES

APPENDIX A - LIST OF COMPANIES

LIST OF TABLES

TABLE		PAGE
II-1	Energy Consumption of Sawmills	4
II-2	Energy Consumption of Pulp Mills	6
II-3	Energy Consumption of Plywood & Veneer Mills	7
II-4	Energy Consumption of Specialty Products	9
II-5	Product Yields for Sawmills	13
II-6	Residue Yields for Pulp and Paper Mills	16
II-7	Product Yields for Softwood Plywood Mills	18
II-8	Product Yields for Waferboard Mills	20
II-9	Product Yields for Particle Board Mills	21
II-10	Product Yields for Red Cedar Shingle Mills	24
II-11	Costs of Electricity in Canada By Province	28 - 31
II-12	Typical Costs for Power From Utility Companies for Two Sizes of Sawmills	32
II-13	Technical Requirements for Parallel Generation of Electricity	34
III-1	Performance Characteristics and Selection Guidelines for Generating Systems	67 - 71
III-2	Compatibility of Candidate Power Systems With Sawmills, Pulp and Paper Mills, and Specialty Product Mills Based on the Electrical to Thermal Energy Ratio	73
III-3	Abridged Summary of Provincial Boiler and Pressure Vessel Acts	75 - 77
IV-1	Typical Operating Performance and Costs for Combustors	80
IV-2	Performance Characteristics of Suspension Burners	86
IV-3	Manufacturers of Suspension Burners	87

LIST OF TABLES (continued)

TABLE		PAGE
IV-4	Performance Characteristics of Multiple Chamber Combustors	92
IV-5	Manufacturers of Multiple Chamber Combustors	93
IV-6	Performance Characteristics of Fluid-Bed Combustors	97
IV-7	Manufacturers of Fluid-Bed Combustors	98
IV-8	Typical Operating Performance and Costs for Gasifiers	103
IV-9	Performance Characteristics of Updraft Gasifiers	108
IV-10	Manufacturers of Updraft Gasifiers	109
IV-11	Data Summary for an Imbert Gasifier With 65 kVa Generator System	115
IV-12	Performance Characteristics of Downdraft Gasifiers	116
IV-13	Manufacturers of Downdraft Gasifiers	117
IV-14	Performance Characteristics of the Standard Solid Fuels Gasifier	120
IV-15	Performance Characteristics of Fluid-Bed Gasifiers	127
IV-16	Manufacturers of Fluid-Bed Gasifiers	128
IV-17	Types and Capacities of Steam Boilers	132
IV-18	Relative Performance Characteristics of Steam Boilers	133
IV-19	Cost of a Field Erected Boiler Plant	137
V-1	Heat Recovery Boiler Design Guidelines	141
VI-1	Effect of Changes in Operating Conditions on Steam Rate	153

LIST OF TABLES (continued)

TABLE		PAGE
VI-2	Typical Steam Rates and Turbine Efficiency for Turbines of Rated Output of Less Than 5 MW	154
VI-3	Typical Turbine Capital and Installation Costs (October, 1981)	159
VI-4	Steam Turbine-Generator Sets Operating and Cost Data	163
VI-5	Proposed Dust Loading Distribution in Gas Turbine Feed	174
VI-6	Manufacturer, Sizes, Costs and Delivery for Turbine/Generator Sets Operating on Natural Gas	175
VI-7	Characteristics of Gas Turbine-Generators by Manufacturer	177
VI-8	Summary of Internal Combustion Engine Manufacturer's Specifications	190
VI-9	Predicted and Actual Test Results of Various Stirling Cycle Engines	196
VI-10	Approximate Power From Reciprocating Engines (E) and Turbines (T) Passing 4,537 kg of Dry Steam Per Hour at Various Inlet and Exhaust Pressures	205
VI-11	Performance Data for A Spilling-Werk Steam Engine	210
VI-12	Performance Data for Fuel Cells	213
VII-1	Capital and Operating Costs for Wood-Fired 250 kW Steam Turbine and Steam Engine Systems	217
VII-2	Costs of 2,500 kW and 5,000 kW Wood-Fired Systems	219
VII-3	Summary of Gasifier/Internal Combustion Engine Systems	223

LIST OF TABLES (continued)

TABLE		PAGE
VII-4	Partial List of Manufacturers of Bottoming Organic Rankine Systems for Power Generation	229

LIST OF FIGURES

FIGURE		PAGE
II-1	Flowsheet of Sawmill Producing Lumber, Pulp Chips and Hog Fuel	11
II-2	Flowsheet of Lumber Finishing Plant Producing Lumber, Pulp Chips, Planer Shavings and Hog Fuel	12
II-3	Flowsheet of Pulp Mill Wood Room Tree-Merchandizer Concept Producing Pulp Chips, Hog Fuel, Peelers and Sawlogs	14
II-4	Flowsheet of Typical Western Softwood Sheathing Plywood Mill Showing Hog Fuel and Sanderdust Sources	17
II-5	Flowsheet of Typical Waferboard Plant Showing Hog Fuel Sources	19
II-6	Flowsheet of Typical Western Red Cedar Shingle Mill Producing Shingles and Hog Fuel	23
II-7	Gross Heat Generation From Residues	25
II-8	Conversion of Thermal Energy to Electricity	26
III-1	Conventional Process Without Residual Energy Recovery	36
III-2	Co-Generation Using a Bottoming Cycle	37
III-3 38	Co-Generation Using a Topping Cycle	
III-4	Co-Generation Using Bottoming and Topping Cycles	39
III-5	Alternative System Configurations	41
III-6	Steam Turbine Topping Rankine Cycles	43
III-7	Approximate Steam Rates For Steam Turbine Topping Cycles	44
III-8	Ratio of Electrical to Thermal Energy for Steam Turbines	46
III-9	Gas Turbine Topping Cycles With a Gasifier	49

LIST OF FIGURES (continued)

FIGURE		PAGE
III-10	Internal Combustion Engine Topping Cycles With a Gasifier	50
III-11	Fuel Cell Topping Cycle With a Gasifier	52
III-12	Indirectly Fired Gas Turbine Topping Cycle With a Combustor	55
III-13	Stirling Engine Topping Cycle With a Combustor	57
III-14	Thermionic Topping Cycle With a Combustor	58
III-15	Direct Fired Gas Turbine Topping Cycle With Fluid-Bed Combustor	62
III-16	Vapour Turbine Bottoming Rankine Cycle Using an Organic Fluid	65
III-17	Approximate Power Output Capacities of Systems for Wood Residue Conversion	74
IV-1	Coen Daz Scroll Feed Burner	82
IV-2	Energex Suspension Burner	83
IV-3	Georgia-Pacific Heat Cell	85
IV-4	Lamb-Cargate Wet Cell	89
IV-5	Anga & Varme Multiple Chamber Combustor	91
IV-6	Energy Products of Idaho Fluid-Bed Combustor	95
IV-7	Updraft Gasifier	99
IV-8	Downdraft Gasifier	100
IV-9	Fluid-Bed Gasifier	102
IV-10	Applied Engineering Co. Gasifier	106
IV-11	Imbert Gasifier Installation at Snowshoe Industries, Fort Providence, NWT	113
IV-12	Standard Solid Fuels Gasifier	119

LIST OF FIGURES (continued)

FIGURE		PAGE
IV-13	Energy Content of Products from ERCO Gasifier	123
IV-14	Omnifuel Gasifier	125
IV-15	Liquid Fuels and Chemicals From Wood by Biological and Thermochemical Processing	129
IV-16	Capital Cost as a Function of Boiler Size and Other Factors	135
V-1	Layouts of Water-Tube Heat Recovery Boilers	139
VI-1	Straight-Through and Extraction Flow Turbines	149
VI-2	Induction and Combination Flow Turbines	150
VI-3	Steam Flow Requirements for a Condensing/Extraction Turbine	156
VI-4	Capital and Installed Costs of Back-Pressure Non-Extraction Turbines	158
VI-5	Steam Flow Requirements for Back-Pressure Turbines	161
VI-6	Approximate Maximum Turbine Inlet Temperatures for Industrial Gas Turbines	166
VI-7	Performance of Simple Cycle Turbine	167
VI-8	Performance of Regenerative Cycle Turbine	169
VI-9	Electrical/Thermal Energy For Simple Cycle Turbine	170
VI-10	Electrical/Thermal Energy for Regenerative Cycle Turbine	171
VI-11	Arrangement of Turbocharged Spark-Ignition Engine	182
VI-12	Arrangement of Turbocharged Dual-Fuel Engine	184
VI-13	Typical Engine Thermal Efficiency at Various Loads	186

LIST OF FIGURES (continued)

FIGURE		PAGE
VI-14	Energy Balance of Internal Combustion Engine	187
VI-15	Electrical/Thermal Energy for Internal Combustion Engines	188
VI-16	Components of a Simple Stirling Engine	194
VI-17	Power and Efficiency Curves for Stirling Engine	198
VI-18	Stirling Engine Thermal Efficiency	199
VI-19	Skinner Steam Engine	203
VI-20	Steam Rates for a Steam Engine at Various Inlet and Exhaust Conditions	206
VI-21	Ratio of Electric to Thermal Energy for Steam Engine	207
VI-22	Fuel Cell Configuration	212
VI-23	Thermionic Converter	215

DEFINITIONS AND NOMENCLATURE

A. DEFINITIONS

$$\text{Compressor Efficiency } (\eta c) = \frac{\text{isentropic work input}}{\text{actual work input}}$$

$$\text{Moisture Content, Dry Basis} = \frac{\text{(weight wet sample-weight dry sample)}}{\text{weight dry sample}} \times 100\%$$

$$\text{Moisture Content, Wet Basis} = \frac{\text{(weight wet sample-weight dry sample)}}{\text{weight wet sample}} \times 100\%$$

Prefixes for Metric Units: $m = 10^{-3}$
$c = 10^{-2}$
$k = 10^{3}$
$M = 10^{6}$
$G = 10^{9}$

$$\text{Specific Work} = \frac{\text{work output}}{\text{mass flow}}$$

$$\text{Turbine Efficiency } (\eta t) = \frac{\text{actual work output}}{\text{isentropic work output}}$$

B. NOMENCLATURE

AHR	Actual Heat Rate, accounting for efficiency of total power cycle (kJ/kWh)
ASR	Actual Steam Rate, allowing for turbine efficiency (kg/kWh)
bf	Board Feet, equal to volume of wood one foot square in area by one inch thick nominal dimensions
BTU	British Thermal Unit
C+	all hydrocarbons with two or more carbon atoms
CH_4	methane
C_2H_6	ethane
CO	carbon monoxide
CO_2	carbon dioxide

$$\text{Cold Gas Efficiency} = \frac{\text{higher heating value of gas at 15°C, 101 kPa}}{\text{higher heating value of input fuel}} \times 100\%$$

dry	dry condition of gas, i.e. no water vapour
E/T	ratio of electrical energy use in GJ to thermal energy use in GJ. Thermal energy referenced to enthalpy at 20°C
g	gram (1 lb = 453.6 g)
h	hour
H_2	hydrogen
HHV	Higher Heating Value for heat of combustion, all water in exhaust as liquid
H_2O	water
HR	Heat Rate. Ratio of energy input or energy removed to work output (kJ/kWh)
J	Joule (1 BTU = 1.055 kJ)
kWh	Kilowatt Hour of energy
l	litre
m	meter (1 ft = 0.3048 m)
M ft^2	1,000 square feet
mmHg	millimeters of mercury column
mo	Month
η_C	compressor efficiency (%)
η_G	generator efficiency (%)
η_T	turbine efficiency (%)
η_{th}	thermal efficiency of engine or turbine (%)
N_2	nitrogen
OD	Oven-Dry, i.e. no moisture content
ODT	Oven-Dry Tons
ODt	Oven-Dry Tonnes

Pa	Pascal (1 psi = 6.895 kPa)
rpm	revolutions per minute
t	tonne (1 ton = 0.907 t)

$$\text{Thermal Efficiency} = \frac{\text{useful work and energy out}}{\text{higher heating value of input fuel}} \times 100\%$$

TSR	Theoretical Steam Rate (kg/kWh)
Vol %	concentration by Volume Percent
W	Watt (1 horsepower = 0.746 kW)
wb	wet basis for moisture content

SUMMARY

A review of options for generating electricity from wood residues was conducted using a unit process approach which enables the assembly of systems from distinct sub-components. These sub-components were primary conversion, secondary conversion and prime movers. Primary conversion includes all equipment such as boilers, combustors and gasifiers that convert wood to useable energy in a primary working fluid. Upgrading of this working fluid or exchange of heat with another working fluid, including heat recovery boilers and hot gas/hot air heat exchangers, was treated as secondary conversion. The working fluid resulting from primary and secondary conversion is the fuel for a prime mover such as a turbine or an internal combustion engine which drives an electrical generator.

A total of nineteen topping and bottoming cycles for generating electricity from wood residues, including proven and emerging technology, were considered. Performance data, operating experience and costs were compiled for equipment included in the unit processes. Subsequently, this detailed data and results of experimental, pilot and commercial operations were used to analyze each of the possible systems for electricity generation.

Table S-1 summarizes the characteristics of the systems that were analyzed. Of all these systems, only those using steam-driven turbines or engines can be considered as proven commercial technology. The combination of a wood gasifier with an internal combustion engine can be considered as near commercial technology. There are numerous suppliers of these systems, most presently developing or demonstrating systems of 100 to 700 kW capacity. Common problems with gasifier/engine systems are poor automatic feeding of residues, tar deposition in the engine, inadequate means of tar disposal and voltage frequency flucuation.

At present no gas turbines are commercially available for operation with wood-derived fuel gas. Problems exist with firing this gas in a turbine because of its low heating value and the presence of particulate matter and tars. Of the several prominent gas turbine companies that were contacted, only Kongsberg reported that some development work was underway to define the turbine modifications needed to accept a fuel such as wood-derived fuel gas. Gas turbines appear suitable only for systems of capacities near the upper part of the capacity range for diesel engines, and larger; roughly capacities greater than 3 MW.

Capital costs were compiled for most systems and system components. For a system of 5 MW, the capital cost of a gasifier/engine combination is very competitive with a boiler/turbine combination. In this case other factors such as availability of proven gasifier systems, operating costs and process demands for steam may strongly influence the choice for system design. For capacities of several hundred kilowatts, a

Table S-1

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
STEAM BOILER:					
-STEAM TURBINE		-proven equipment with many options -good operating flexibility -worker familiarity -moderate cost -packaged units available	-requires trained operators -need water source -need use for steam -high maintenance	-none	-Pulp and paper mills -Sawmills with dry kilns -Plywood and veneer mills
SYSTEM STATUS	-commercial				
POWER OUTPUT (kW)	->300; >1,500 typical				
E/T	-0.04-0.45				
BOILER CODE	-applicable				
-STEAM ENGINE		-lower steam rates than turbines below 2 MW -good for variable speed and slow speed applications -good torque characteristics -low maintenance -worker familiarity -good operating flexibility like turbines	-limited availability, sizes and options -requires trained operators -need water source -need use for steam	-steam engine	-Pulp and paper mills -Sawmills with dry kilns -Plywood and veneer mills
SYSTEM STATUS	-commercial				
POWER OUTPUT (kW)	-25 - 3,000; up to 20,000 possible				
E/T	-0.04 - 0.45				
BOILER CODE	-applicable				
GASIFIER AND PYROLYZER:					
-FUEL CELL		-no moving parts in fuel cell -low emissions -operating flexibility -modular construction -high efficiency, over-wide load range	-unproven -no commercial systems available -very sophisticated unfamiliar technology -high cost	-gasifier -gas reformer -fuel cell	-Unknown
SYSTEM STATUS	-conceptual (pilot operating on naptha)				
POWER OUTPUT (kW)	->20				
E/T	-1 - 1.2				
BOILER CODE	-not applicable				

Table S-1 (continued)

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
GASIFIER AND PYROLYZER:					
-GAS TURBINE		-suitable for automation -compact -low labour requirement	-best for base load applications -poor part-load performance -operational inflexibility -fouling of turbine -unproven, limited availability -high maintenance	-gas cleanup -turbine combustor -control system	-possibly sawmills -specialty products mills
SYSTEM STATUS	-conceptual/ research				
POWER OUTPUT (kW)	-500 - >100,000				
E/T	-0.2 - 0.8				
BOILER CODE	-not applicable				
-INTERNAL COMBUSTION ENGINE		-suitable for automation -low labour requirement -moderate maintenance -fair equipment availability	-possible problem with engine control due to fluctuations of gas quality -tar and particulate deposits in engine -scrubber water disposal -lube oil contamination	-gas cleanup -control system	-sawmills -specialty products mills
SYSTEM STATUS	-demonstration/ commercial				
POWER OUTPUT (kW)	-100 - >2,000				
E/T	-0.5 - 0.9				
BOILER CODE	-not applicable				
-STEAM TURBINE		-tars incinerated, maximizing energy recovery from fuel gas -gas available for other uses -others as per steam boiler	-limited equipment availability -more complex than combustor/boiler combination -others as per steam boiler	-gas burner -steam engine	-pulp and paper mills -sawmills with dry kilns -plywood and veneer mills
SYSTEM STATUS	-demonstration/ commercial				
POWER OUTPUT (kW)	-300 - 8,000				
E/T	-0.4 - 0.45				
BOILER CODE	-applicable				
-STEAM ENGINE					
SYSTEM STATUS	-demonstration commercial				
POWER OUTPUT (kW)	-25 - 3,000				
E/T	-0.04 - 0.45				
BOILER CODE	-applicable				

Table S-1 (continued)

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
COMBUSTOR:					
-GAS TURBINE		-clean gas for turbine -suitable for automation -if closed cycle, good part load performance and efficiency -low to moderate turbine maintenance	-need use for hot gas or steam -thermal inertia poses control problem -best for base load, steady operation -limited availability of heat exchanger for high temeperature	-heat exchanger -control system	-pulp and paper mills
SYSTEM STATUS	-pilot				
POWER OUTPUT (kW)	->500				
E/T	-approx. 0.08				
BOILER CODE	-not applicable; with combined cycle is applicable				
-STIRLING ENGINE		-engine uses isolated working fluid -long engine life -suitable for automation	-ash fouling of heater in stirling engine -thermal inertia poses control problem -worker infamiliarity -limited availability of engine -small output	-gas cleanup -Stirling engine	-sawmills -specialty products mills
SYSTEM STATUS	-prototype/ demonstration				
POWER OUTPUT (kW)	-<40, presently				
E/T	-0.6 - 1.0				
BOILER CODE	-not applicable				
-STEAM TURBINE		-hot gases available for other uses -flexible modular system structure -others as per steam boiler	-thermal inertia poses control problem -others as per steam boiler	-steam engine	-pulp and paper mills -sawmills -plywoos and veneer mills
SYSTEM STATUS	-commercial				
POWER OUTPUT (kW)	-300 - 11,000				
E/T	-0.04 0.45				
BOILER CODE	-applicable				
-STEAM ENGINE					
SYSTEM STATUS	-demonstration/ commercial				
POWER OUTPUT (kW)	-25 - 3,000				
E/T	-0.04 - 0.45				
BOILER CODE	-applicable				

Table S-1 (continued)

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
COMBUSTOR:					
-**THERMIONICS**		-no moving parts in converter -good for combined cycle	-unproven -no commercial systems available -high cost -unfamiliar technology -need use for hot gases	-all system components	-unknown, E/T poor at present for all mills
SYSTEM STATUS	-research				
POWER OUTPUT (kW)	-4 - >100,000				
E/T	-0.02, presently, 0.2 - 0.25, future				
BOILER CODE	-not applicable				
PRESSURIZED COMBUSTOR:					
-**GAS TURBINE**		-suitable for automation -compact	-thermal inertia of combustor poses control problem -best for base load -ash fouling and wear of turbine -no commercial systems available	-gas cleanup -gas turbine -fuel feed system	-sawmills -specialty products mills
SYSTEM STATUS	-research				
POWER OUTPUT (kW)	-1,000 tested				
E/T	-0.2 - 0.8				
BOILER CODE	-not applicable				
PULVERIZER:					
-**SOLID FUEL TURBINE**		-simple system -suitable for automation	-high power required for pulverizing -ash fouling and wear of turbine -no commercial systems available	-all system components	-sawmills -specialty products mills
SYSTEM STATUS	-conceptual				
POWER OUTPUT (kW)	-unknown				
E/T	-0.2 - 0.8				
BOILER CODE	-not applicable				

Table S-1 (continued)

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
LIQUEFACTION:					
-GAS TURBINE, INTERNAL COMBUSTION ENGINE		-conventional prime movers	-unproven -very low conversion efficiency -high cost	-all system components	-unknown
SYSTEM STATUS	-conceptual				
POWER OUTPUT (kW)	-N/A				
E/T	-N/A				
BOILER CODE	-not applicable				
HEAT RECOVERY BOILER:					
-ORGANIC VAPOR TURBINE		-recovery of waste heat -higher efficiency than steam below 400°C -best for recovery at 100°C - 300°C	-toxic fluids -problem to handle and dispose of fluids -limited availability of systems -degradation of fluids -worker unfamiliarity	-all system components	-pulp and paper mills
SYSTEM STATUS	-demonstration				
POWER OUTPUT (kW)	-0.1 - 1,000				
E/T	-does not apply				
BOILER CODE	-probably does apply				
-STEAM TURBINE		-SEE STEAM BOILER			
SYSTEM STATUS	-commercial				
-STEAM ENGINE					
SYSTEM STATUS	-commercial				

gasifier/engine system has a capital cost advantage over an alternate steam-based system.

A promising emerging system for electricity generation consists of a combustor or gasifier combined with a Stirling engine. Development of these systems is presently hampered by the 40 kW upper limit on size of available Stirling engines. A study of this type of system is presently being conducted by United Stirling Inc. in the United States.

I INTRODUCTION

A. Background

Many forest products industries are in a position to self-generate their thermal energy and electrical needs with mill residues. Historically, the pulp and paper industry has been a major user of wood residues for generation of process steam. Other industries have utilized these residues for heating and drying applications. For economic reasons, only industries with substantial electricity requirements, primarily the pulp and paper industry, have utilized wood residues to generate electricity. Other industries have relied on provincial utilities for power.

Electricity generating facilities in pulp and paper mills may range from as low as 2 MW to upwards of 30 MW for a 1,000 t/d Kraft pulp mill, and as high as 70 MW for an integrated paper mill of 1,300 t/d capacity. Having large requirements for process heat and steam, pulp and paper mills are well suited to conventional steam cycle power generating systems. Other types of forest products mills have lower requirements for electricity and often negligible requirements for steam. For these mills, the conventional steam cycle is often unattractive. Power generating systems besides the steam cycle are becoming commercially available, but often are not widely recognized or understood.

This report addresses the task of reviewing the technology for generating electricity from wood residue fuels. The systems of primary interest are those having sizes below what is recognized as being most appropriate for steam cycles. This range of interest has therefore been set at below 5 MW.

Because of the demand for both thermal and electrical energy in most mills, the systems considered in this study were treated as cogeneration systems, although electricity was assumed to be the primary energy product.

A tremendous diversity exists for the configurations and types of equipment for electricity generating systems. To avoid redundant treatment of similar systems or undue emphasis of some systems over others, and to allow the reader to assemble systems to match the needs of a particular set of circumstances, the review of the systems is presented using a unit processes approach. The unit processes chosen as building blocks of the system are the Primary Conversion Unit, the Secondary Conversion Unit and the Prime Mover Unit, these covering the transformation of raw wood fuel to electricity and thermal energy. Performance, cost and supply information was compiled for all the distinct equipment within each of the unit processes.

B. Scope

This study concerned itself primarily with systems capable of generating less than 5 MW of power. Larger installations were not excluded, but it was felt that existing steam boiler/turbogenerator designs are so well established for integrated mills that there was little merit in covering them in much detail, so most emphasis was placed on smaller systems.

The major objectives of the study were:

1. Review and evaluate systems available for generating electricity from wood residues in sizes typically less than 5 MW;

2. Compile performance and cost data for the components of generating systems so that it may be applied to most potential installations;

3. Consider proven systems as well as those in a research or demonstration stage;

4. Develop a unit process format which the forest industry can use for selecting systems and that is amendable to revision when new data is available.

C. Format

In order to attain maximum benefit from this report, it is important to fully understand its format and how it is meant to be used. Each complete electricity-generating system discussed in this report consists of distinct subsystems or unit processes. These unit processes fall into three categories: Wood Conversion Units, Power Fluid Heat Exchange Units, and Prime Mover Units. Each complete system consists of at least one Wood Conversion unit and one Prime Mover unit. In some cases, the system consists of more than one of the preceding units as well as a Power Fluid Heat Exchange unit.

Chapter II of the report provides background information regarding energy requirements and mill residue production for various types of mills in the forest products industry.

Chapter III discusses, in general terms, the various complete systems identified as being available to produce electricity from wood residue. From these sections, it is hoped that the reader may determine which complete systems suit his own particular operation and requirements.

Having determined complete systems of interest and given information as to which unit processes constitute these systems from Chapter II, the reader may find information regarding specific unit processes in Chapters IV, V, and VI. These sections provide details which include operating data, typical costs, system status, and a description of the equipment available from representative manufacturers.

In Chapter VII, companies which supply complete turnkey electricity-generating systems are discussed. Details are given pertaining to overall system operational data, costs, and where available, a list of installations.

Finally, in Chapter VIII a discussion is presented regarding the status of development of the various complete systems.

Using this format, it is hoped that all available options for the reader are covered, including both complete systems and unit processes which may be added to equipment already operating at a plant.

II POTENTIAL FOR ELECTRICITY GENERATION FROM WOOD RESIDUES

Before any systematic analysis of possible alternatives for electricity generation can proceed, a thorough study of the total energy requirements in the form of thermal and electrical energy is essential. This study must determine the demand and the energy consumption during normal operation as well as during peaking situations such as start up of process equipment. If an electricity system must cogenerate heat and electricity, then data for the interrelation of these two energy demands must also be known. These data should give an understanding of the range in value of the ratio of electrical to thermal energy during operation and whether there is a relationship in the timing of the demand for each energy form.

In numerous circumstances, reliance on a single cogeneration system for total energy needs may not be possible because of the system's performance characteristics or may not be practical for reasons of layout, energy forms, grade of energy or cost. Connection to alternate sources of electricity is a common practice to supply backup power in case of system failure as well as to have a satisfactory supply of power during high-demand situations.

This chapter presents data and discussion to allow: 1) an estimation of the energy requirements of most types of forest products mills; 2) an estimation of the residues available at these mills for conversion to thermal and electrical energy; and 3) a consideration of the consequences of interfacing an in-plant generating facility with a provincial utility. The data for energy requirements covers sawmills, plywood and veneer mills, pulp and paper mills, and specialty products mills such as those for production of shakes, shingles and particle board. These data are industrial averages for thermal and electrical energy requirements and cannot supplant a site-specific analysis. For each category of forest products mill, data are presented for the average yield of residue per unit of primary product, where possible in Eastern and Western Canada. These yield data are converted to indicate the electrical energy which could be produced practically with commercial systems.

The discussion of competing sources of electricity presents data on the cost of power obtained from provincial utilities and what policy and cost structure exists for the interconnection of a mill power plant to the utility grid and the sale and purchase back of power by the utility.

A. Energy Requirements of Mills

1. Sawmills

Figures from five sources (Table II-1) indicate that the electrical energy consumption rate for lumber mills ranges from

Table II-1

ENERGY CONSUMPTION OF SAWMILLS

Source	Electrical Energy Consumption, E (kWh/m^3)	Thermal Energy Consumption, T (GJ/m^3)	Total[a] Energy Consumption (GJ/m^3)	E/T[a] (GJ/GJ)
Statistics Canada[b]				
Newfoundland	31.5			
Nova Scotia	67.5			
New Brunswick	83.3			
Quebec	76.1	N/A	N/A	N/A
Ontario	64.4			
Manitoba	30.6			
Saskatchewan	44.7			
British Columbia	54.7			
Total Canada	60.0			
B.H. Levelton & Associates Ltd.[c]				
Mill A	43.3	0.734	0.890	0.212
Mill B	83.9	0.280	0.582	1.080
Mill C	58.9	0.463	0.675	0.459
Mill D	76.3	0.396	0.671	0.694
Koch Softwood Lumber	88.7	1.648	1.968	0.194
Neill & Gunter[d]				
Mill A	51.2	1.515	1.699	0.122
Mill B	49.4	1.153	1.331	0.154
Mill C	23.4	0.751	0.835	0.113
Mill D	59.1	0.563	0.776	0.377
Mill E	76.1	0.635	0.909	0.432
B.C. Utilities Commission B.C. Total	58.3	1.53	1.74	0.137

[a] 1 kWh = 3.6×10^{-3} GJ.
[b] Purchased electricity only. Does not include in-house power generation.
[c] Alberta.
[d] Eastern Canada.

approximately 20 to 90 kWh/m^3. The consumption rate is dependent on lumber species, mill size and age, and equipment used in the mill.

Thermal energy consumption rates range from 0.3 to 1.6 GJ/m^3. The thermal energy consumption rate is dependent on the volume of lumber that is kiln dried and its initial and final moisture content in addition to the influencing variables indicated for electricity consumption.

2. Pulp and Paper Mills

Table II-2 gives four sets of figures for the energy consumption in the Pulp and Paper industry. The different figures for the electricity consumption rates for B.C. mills can be accounted for in that the B.C.U.C. report included in-house electricity generation whereas Statistics Canada figures do not.

The energy consumption rate for pulp and paper mills depend on mill size, end product and process type. This is readily shown in the Ontario Hydro Report figures for 1974.

3. Plywood and Veneer Mills

Data from four sources (Table II-3) give a range in the electricity consumption rate of 28.4 to 205.3 kWh/m^3 and in the thermal energy consumption rate of 3.03 to 7.06 GJ/m^3. In addition to reasons given in the previous section, these energy consumption rates are dependent upon the board thickness.

4. Specialty Products Mills

Specialty wood products manufactured in Canada are, for the most part, comprised of three types of products: shakes and shingles; composite fibreboards; and finished construction materials.

The largest electrical energy consumer of these specialty wood product plants is the finished construction material manufacturers whose products include sash and doors, cabinets and cupboards, roof trusses, pre-fabricated houses, etc. The annual electrical energy purchases by these plants in Canada during 1978 amounted to 242,300 MWh. Due to the nature and diversity of this end product, thermal energy consumption and energy consumption per unit product are difficult to determine.

The next largest user of energy in the specialty wood products field is composite fibreboard plants. These end products include

Table II-2

ENERGY CONSUMPTION OF PULP MILLS

Source	Electrical Energy Consumption, E (kWh/t)	Thermal Energy Consumption, T (GJ/t)	Total [a] Energy Consumption (GJ/t)	E/T [a] (GJ/GJ)
Statistics Canada [b]				
Quebec	1640	N/A	N/A	N/A
Ontario	1011			
British Columbia	675			
Total Canada	1089			
B.C. Utilities Commission B.C. Total	861.1	26.0	29.1	0.119
Zerbe	N/A	N/A	36.4 - 47.9	
Ontario Hydro Report (1974)				
Newsprint	1700 - 2200	13.6 - 17.7	19.7 - 25.6	0.448
Kraft	880 - 1210	19.3 - 27.3	22.5 - 31.7	0.163
Sulfite	550 - 715	16.1 - 22.4	18.1 - 25.0	0.120
Paperboard	880 - 1320	14.4 - 19.3	17.6 - 24.1	0.236

[a] 1 kWh = 3.6×10^{-3} GJ.

[b] Includes purchased electricity only. Does not include in-house power generation.

Table II-3

ENERGY CONSUMPTION OF PLYWOOD & VENEER MILLS

Source	Electrical Energy Consumption, E (kWh/m^3)	Thermal Energy Consumption, T (GJ/m^3)	Total [a] Energy Consumption (GJ/m^3)	E/T [a] (GJ/GJ)
B.C. Utilities Commission				
B.C. Total	133.3	3.03	3.51	0.158
Zerbe	N/A	N/A	3.30	
Koch				
Softwood Plywood	28.4	4.75	4.85	0.021
Hardwood Plywood	47.9	7.06	7.23	0.024
Neill & Gunter (b)				
Mill A	168.5	5.26	5.87	0.116
Mill B	205.3	5.70	6.44	0.129
Mill C	190.3	5.69	6.37	0.121

[a] 1 kWh = 3.6 X 10^{-3} GJ.

[b] Eastern Canada.

particle board, insulation board, hardboard and medium-density fibreboard, all of which differ by size of material used in its manufacture and by final density. Energy consumption rates for these products range anywhere from 153 kWh/m^3 to 3079 kWh/m^3 for electrical energy and 1.35 GJ/m^3 to 10.89 GJ/m^3 for thermal energy (Table II-4).

The least significant energy consumers of the specialty wood product mills are the shake and shingle producers. In Canada 90% of the total production of shakes and shingles is in the province of British Columbia. Figures given by the British Columbia Utilities Commission Report (Table II-4) give an electricity consumption rate of 69.4 kWh/m^3 and a thermal consumption rate of 0.94 GJ/m^3.

B. Meeting Electricity Demands With Mill Residues

Previously, Section A discussed the demand for thermal and electrical energy in most types of wood processing mills. These data illustrate the dependence of the proportional split of thermal and electrical energy on the type of mill. The energy use per unit of product also varys widely between different mills for the same product, this being due to site specific factors such as age of mill equipment, mill location, type of processing, etc.

In assessing the viability of generating electricity at a mill using produced or purchased residues, it is essential that the demand for thermal and electrical energy be known under both normal and peak production conditions for the mill in question.

This section presents typical flowsheets for sawmills, pulp and paper mills, plywood mills, waferboard mills and shingle mills. These are generalized flowsheets indicating primary process units and flows of primary and secondary products, as well as of residues that are potential fuels.

To complement the flowsheets, data were compiled to indicate the yield of products and residues for each of the mill types. These product yields can be used to estimate the quantity of residues produced in conjunction with the primary mill product. More accurate estimates of residue production may readily be determined by site-specific material balances.

This section concludes with a discussion of the quantity of residues needed to supply the thermal and electrical energy demands of example mills.

Table II-4

ENERGY CONSUMPTION OF SPECIALTY PRODUCTS MILLS

Source and Product	Electrical Energy Consumption, E (kWh/m^3)	Thermal Energy Consumption, T (GJ/m^3)	Total [a] Energy Consumption (GJ/m^3)	E/T [a] (GJ/GJ)
B.C. Utilities Commission				
Particle Board	152.8	1.35	1.90	0.407
Shakes & Shingles	69.4	0.94	1.19	0.266
Statistics Canada [b]				
Particle Board	166.7	---	---	---
Koch				
Particle Board [c]	323.7	2.61	3.77	0.447
Insulation Board [d]	509.0	2.09	3.93	0.876
Wet Formed Hardboard [e]	3078.6	10.89	21.97	1.018
Medium Density Fibreboard [g]	872.5	4.66	7.80	0.675
Oak Flooring [f]	163.5	3.38	3.96	0.174

[a] 1 kWh = 3.6×10^{-3} GJ.
[b] Purchased electricity only. Does not include in-house power generation.
[c] Based on 400 kg/m^3 density.
[d] Based on 320 kg/m^3 density.
[e] Based on 961 kg/m^3 density.
[f] Based on 601 kg/m^3 denisty.
[g] Based on 721 kg/m^3 density.

1. Sawmills

Flowsheets for sawmills producing rough green lumber and dressed green or dry lumber are shown in Figures II-1 and II-2, respectively. These indicate the primary lumber product, the secondary pulp chip product and the dry and wet residues. Hog fuel represents the various mixtures of dry and wet residues after a final size reduction in a hog. For simplicity, no effort was made to separate the dry and wet residues in the flowsheet, since such details of the handling system will depend strongly on the facilities in place at the mill. In many cases it will be very beneficial to separate the dry and wet residues and to utilize them in appropriate conversion systems.

Product yield data obtained from a survey of the literature are presented in Table II-5. These data are primarily for softwoods, the only data for hardwoods being available from the report by Huffman. Considering the differences in the locations of the sawmills, the yields of bark, sawdust and shavings are quite consistent. There is a substantial variation of the yield of chips for the sawmills, as is to be expected because of differing local markets.

The average yield data indicate that about 113 OD kg of bark, 96 OD kg sawdust and 77 OD kg of shavings are produced with each cubic meter of lumber. Of these residues the total bark and sawdust, 209 kg/m^3, is the primary residual fuel. The 77 OD kg/m^3 of shavings may be used for fuel if other more attractive markets do not exist.

An example of the rate of residue production can be developed by assuming a softwood lumber production rate of 236 m^3/day (100 Mbf/day). In this case, 26.7 OD t of bark and 22.7 OD t of sawdust are produced, resulting in a fuel supply of 49.4 OD t per day, or 2.1 OD t/h. A quick means of translating this production rate into equivalent gross heat generation and electricity generation are given in Section B-5.

2. Pulp and Paper Mills

Wood residues suitable for fuel are produced from the wood room of pulp and paper mills. These residues (Figure II-3) consist of sawdust and lilypads from cut-off saws, bark and chip fines. In total, these residues form the hog fuel supply.

No data were located for the production of sawdust and other unchipped waste from the cut-off saws. Data for the quantity of bark and chip fines produced were obtained from reports by Dobie and Wright and the British Columbia Wood-Waste Energy Coordinating

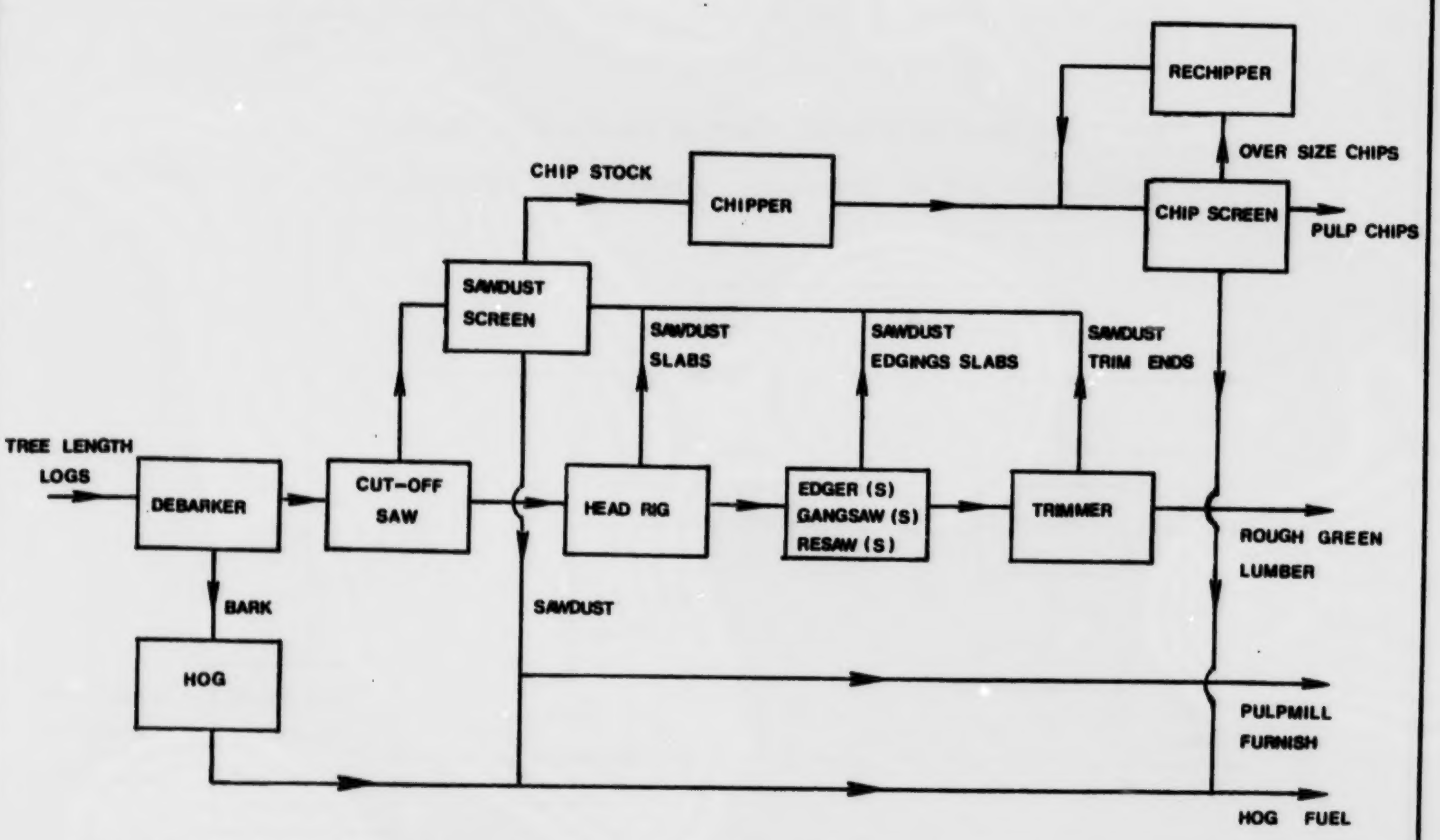


FIGURE II-1 FLOWSHEET OF SAWMILL PRODUCING LUMBER, PULPCHIPS AND HOG FUEL

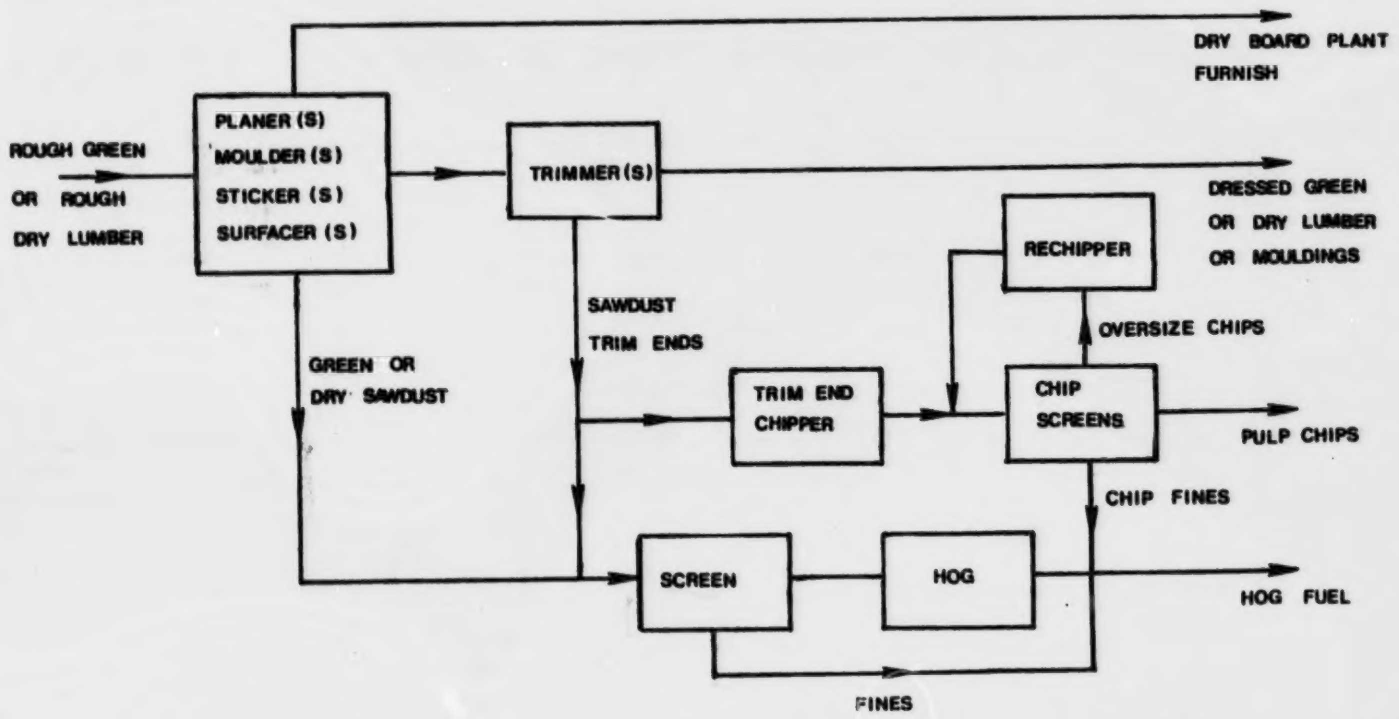


FIGURE II-2 FLOWSHEET OF LUMBER FINISHING PLANT PRODUCING LUMBER, PULP CHIPS, PLANER SHAVINGS AND HOG FUEL

Table II-5

PRODUCT YIELDS FOR SAWMILLS

ITEM	PRODUCT YIELD OD kg/m^3 LUMBER (ODT/Mfbm)							
Species Reference	Softwoods Moore	Softwoods Corder	Softwoods Corder	Softwoods Neil & Gunter	Softwoods Dobie & Wright	Softwoods Huffman	Red Oak Huffman	Softwoods Average of Data For Softwoods
Location	Oregon & California	West Oregon	East Oregon	Eastern Canada	British Columbia	Eastern Canada	Eastern Canada	
Residue Type: Bark	150 (0.39)*	110 (0.285)**	88 (0.228)**	127 (0.330)	108 (0.280)	96 (0.25)	257 (0.669)	113 (0.294)
Sawdust	82 (0.213)	114 (0.297)	102 (0.264)	108 (0.280)	89 (0.231)	81 (0.21)	232 (0.604)	96 (0.249)
Shavings	65 (0.170)	83 (0.216)	74 (0.192)	88 (0.230)	79 (0.206)	65 (0.17)	N/A	77 (0.197)
Total	297 (0.773)	307 (0.798)	264 (0.684)	323 (0.840)	276 (0.717)	242 (0.63)	489 (1.273)	286 (0.740)
Coarse Residues (as chips)	156 (0.405)	223 (0.580)	198 (0.516)	434 (1.13)	202 (0.525)	350 (0.91)	375 (0.976)	
Lumber		338 (0.878)	300 (0.780)	330 (0.859)		277 (0.72)		

* Density of Bark - 192 ODkg/m^3; Sawdust - 144 ODkg/m^3; Shavings - 96 ODkg/m^3
Density of Chips - 160 ODkg/m^3

** Density of Bark - 0.365 m^{3}SWE/m^3; Sawdust - 0.36 m^{3}SWE/m^3; Shavings 0.225m^{3}SWE/m^3
Density of Chips - 0.4 m^{3}SWE/m^3

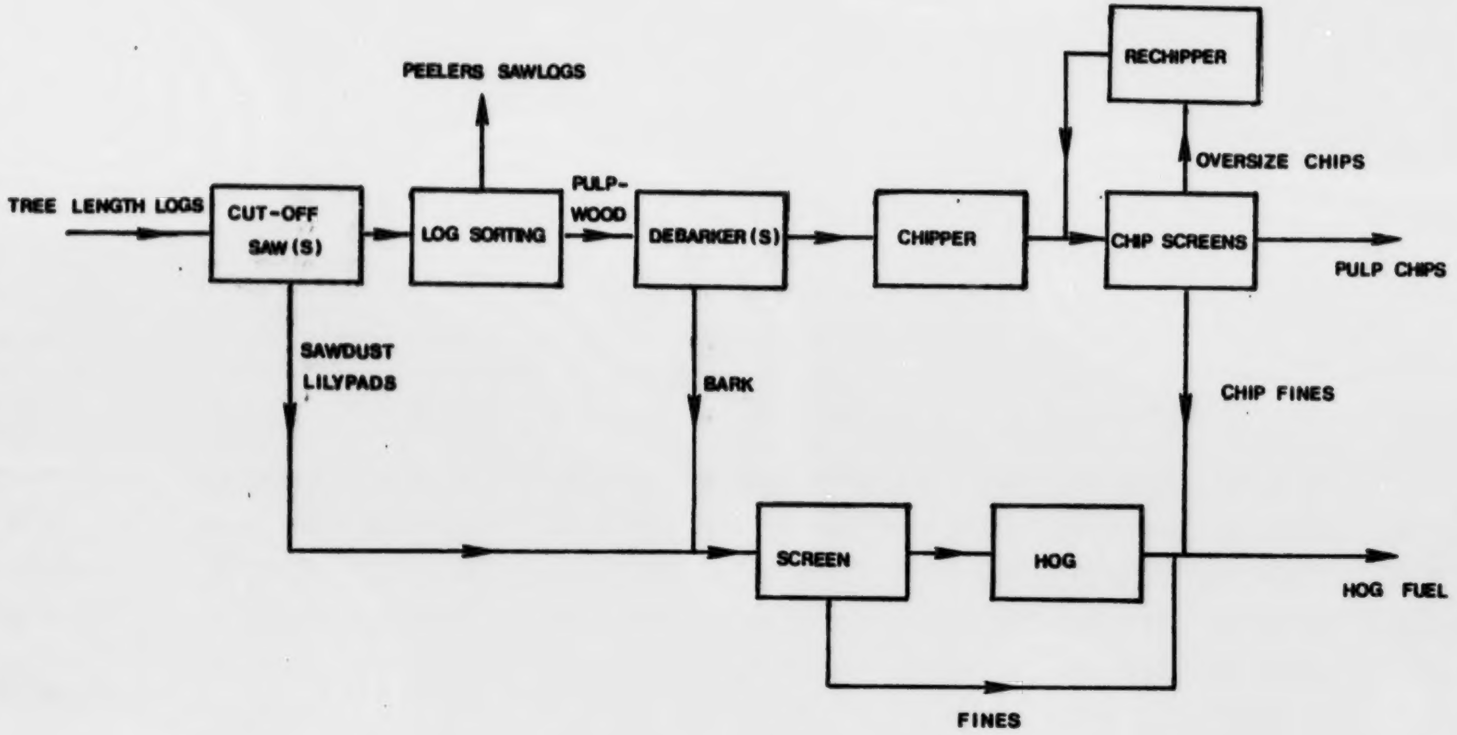


FIGURE II-3 FLOWSHEET OF PULP MILL WOOD ROOM TREE-MERCHANDIZER CONCEPT PRODUCING PULP CHIPS, HOG FUEL, PEELERS AND SAWLOGS

Committe (Table II-6). The data for chip fines production are in units of weight percentage of consumption for a pulpmill.

Assuming a greenwood density of 400 OD kg/m^3, the bark production from spruce (Table II-6) is 0.168 OD kg/OD kg wood. When added to 0.05 OD kg fines/OD kg wood chipped the total wood residue production rate becomes 0.218 OD kg/OD kg of wood chipped and screened.

3. Plywood and Veneer Mills

Figure II-4 presents a schematic of a sheathing plywood plant where wet and dry hog fuel are segregated. Sanderdust is also produced with the finished sheathing.

Product yields for the United States and Eastern Canada are shown in Table II-7. The averaged data indicates about 122 OD kg of bark and 55 OD kg of sawdust and sanderdust are produced with each cubic meter of 9.5 mm (3/8 inch) thick plywood, for a total of 177 OD kg/m^3, 9.5 mm thick plywood.

4. Specialty Products Mills

Mills considered in this section produce waferboard, particle board and cedar shingles. These were selected because of their significance in the forest products industy and are not meant to be a comprehensive survey of mills outside those previously discussed. The flowsheets and product yield data illustrate the procedure for estimating residue production available for conversion to thermal and/or electrical energy.

Material flows and yields of products are presented for a typical waferboard mill in Figure II-5 and Table II-8, respectively. Dry fine fuels are produced from exhaust cyclones, the plant vacuum system and product trim, while residues upstream of the drier are segregated to produce a wet hog fuel stream. The total yield of bark and sawdust is about 397 OD kg/m^3, 12.7 mm board. This result may be applied directly to other board thicknesses without conversion.

Because of the similarity of the production steps for particle board to those for waferboard, only residue yields are presented (Table II-9). These data show that from 52.9 to 127.9 OD kg of dry residues in the form of trim waste and sanderdust are produced per cubic meter of particle board. The assumed furnish for this operation is a mixture of planer shavings, sawdust and plywood trim.

Figure II-6 presents a flowsheet for a typical western red cedar shingle mill. Product yield data (Table II-10) apply to a

Table II-6

RESIDUE YIELDS FOR PULP AND PAPER MILLS

ITEM	RESIDUE YIELD	
Location	Vancouver Forest District	Prince George Forest District
Primary Species	Western Hemlock	Spruce
Bark*		
(m^3/m^3 wood)	0.111	0.133
(OD kg/m^3 wood)**	56	67
Chip Fines		
(wt% of consumption)	5 (estimated)	4 - 6

* Volumes and weights given at mill allowing for losses during harvesting and delivery.

** Bark density = 500 OD kg/m^3 (Dobie and Wright)

Source: B.C. Wood-Waste Energy Coordinating Committee

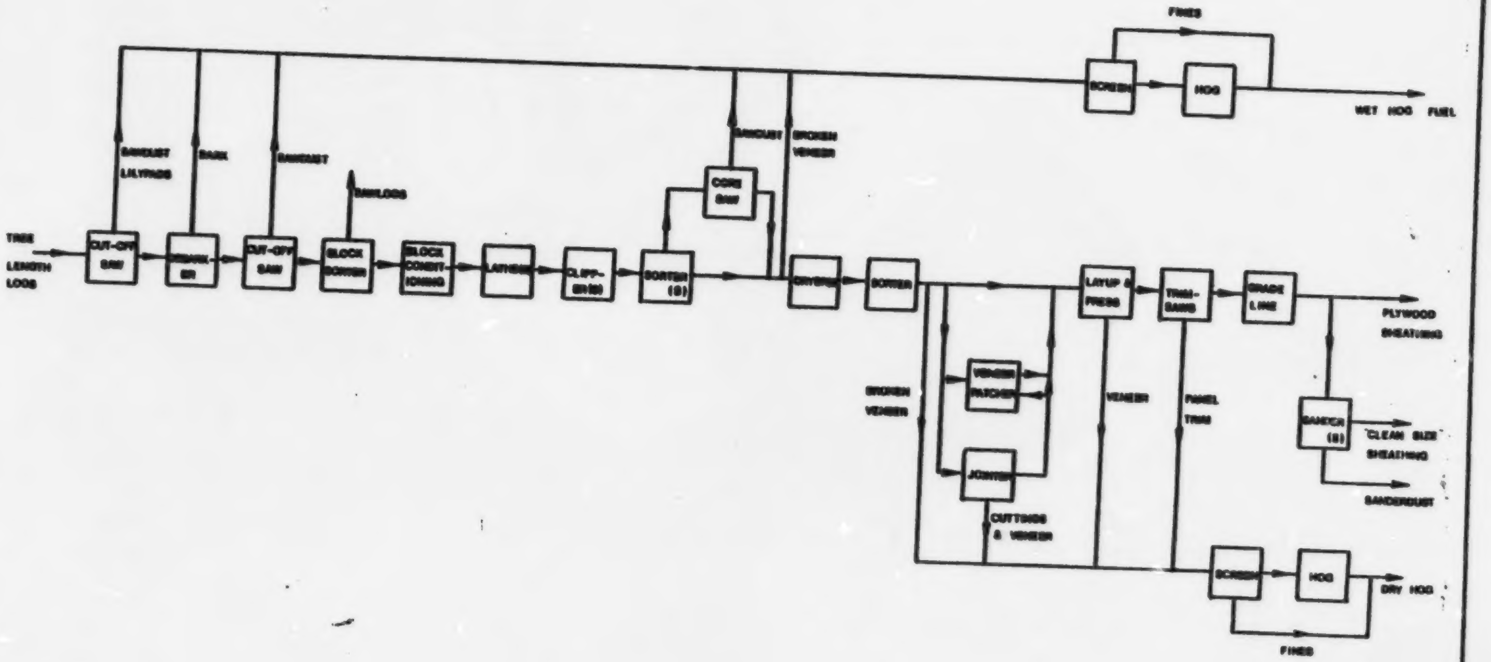


FIGURE II-4 FLOWSHEET OF TYPICAL WESTERN SOFTWOOD SHEATHING PLYWOOD MILL SHOWING HOG FUEL AND SANDERDUST SOURCES

Table II-7

PRODUCT YIELDS FOR SOFTWOOD PLYWOOD MILLS

ITEM	PRODUCT YIELD OD kg/m^3, 9.5 mm (ODT/M ft^2 3/8" PLYWOOD)			
Reference	Corder	Koch	Neill & Gunter	
Location	Oregon	United States	Eastern Canada	Average of All Data
Residues:				
Bark	135 (0.132)	109 (0.106)	123 (0.12)	122 (0.119)
Log trim	47 (0.046)			
Core	51 (0.050)	120 (0.117)	119 (0.116)	
Green Clippings, Roundup & Spurtrim	256 (0.250)	283 (0.276)		
Dry Veneer Loss & Panel Trim	90 (0.088)	76 (0.074)		
Sawdust & Sanderdust	22 (0.021)	22 (0.021)	123 (0.12)	55 (0.054)
Total	601 (0.455)	610 (0.594)	---	---
Plywood	475 (0.463)	490 (0.478)*		

* Based on shipping weight given by Dobie & Wright and assumed moisture content of 15%, wb.

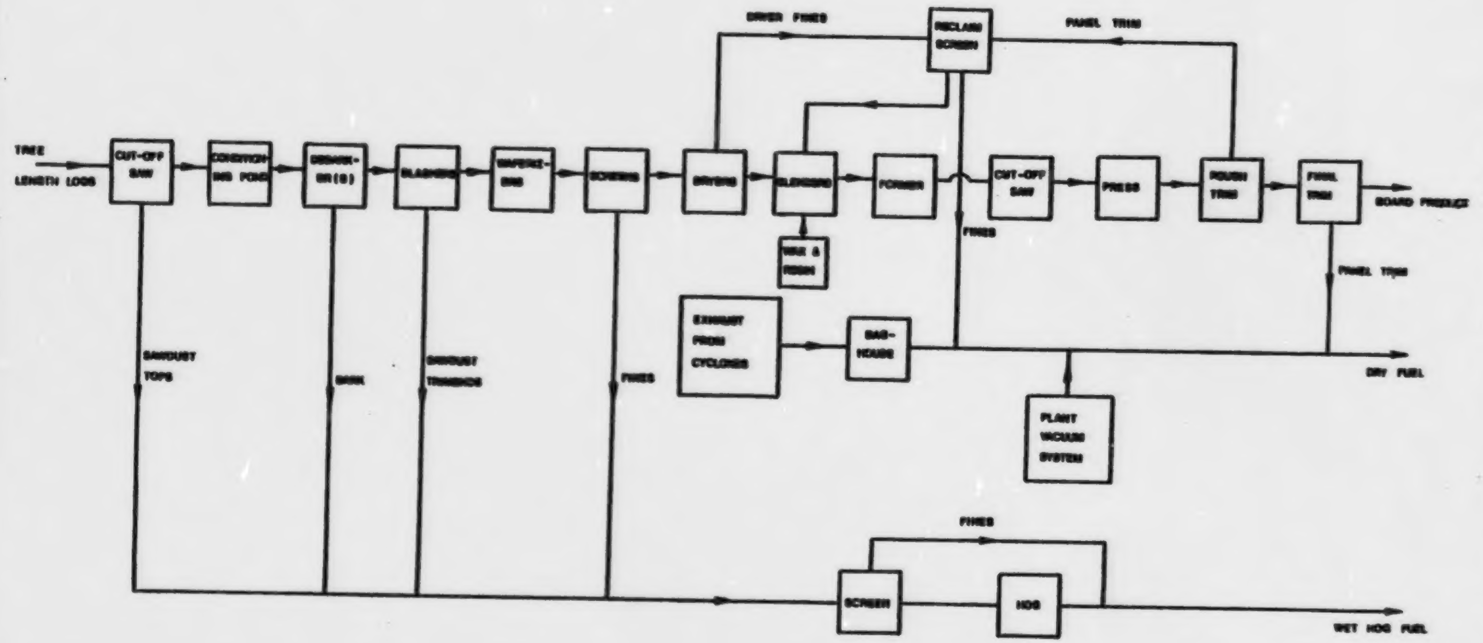


FIGURE II-5 FLOWSHEET OF TYPICAL WAFERBOARD PLANT SHOWING HOG FUEL SOURCES

Table II-8

PRODUCT YIELDS FOR WAFERBOARD MILLS

ITEM	PRODUCT YIELD OD kg/m^3, 12.7 mm BOARD (ODT/M ft^2 1/2" BOARD)
Reference	Koch
Location	United States
Residues:	
Bark	180 (0.234)*
Sawdust Trim & Rejects	217 (0.282)
Total	397 (0.516)
Additives	44 (0.057)
Lumber	812 (1.056)

* Assumes 641 kg/m^3 (0.831 ton/ft^2, 1/2" board) for product density.

Table II-9

PRODUCT YIELDS FOR PARTICLE BOARD MILLS

ITEM	PRODUCT YIELD OD kg/m^3, 15.9 mm BOARD (ODT/M ft^2 5/8" BOARD)	
Reference	Koch	Dobie & Wight
Location	United States	British Columbia
Residues:		
Trim Waste	6.8 (0.011)	58.4 (0.095)
Sander Dust	46.1 (0.075)	69.5 (0.113)
Total	52.9 (0.086)	127.9 (0.208)
Additives	42.4 (0.069)	24.0 (0.039)
5/8" Board	480 (0.781)	480 (0.781)

NOTE: -Board specifications: Density = 481 kg/m^3; 1.28 M ft^2/ton 5/8" board

-Density of input wood = 384 kg/m^3

red cedar shingle mill producing a Fivex shingle (Dobie and Wright). The total residue amounts to 1.488 OD kg/OD kg shingles, or about 0.061 ODT/square.

5. Thermal and Electrical Energy Generation From Residues

Knowing the residue production rate of a mill, the gross heat generation rate that is possible can be estimated from Figure II-7. These curves cover the range of heating values typical for wood and bark.

To arrive at the net heat generation rate, such as the output from a conventional boiler, the gross heat generation rate must be multiplied by the appropriate thermal conversion efficiency of the system. For example, a mill producing 5 OD t/h of residuals for fuel can generate 99 GJ/h of gross thermal energy and, with a boiler operated on wet fuel at 65% conversion efficiency, 64 GJ/h of net thermal energy if the fuel has a heating value of 19.8 MJ/kg, dry basis.

The quantity of electricity generated from wood residues depends on the overall conversion efficiency of the system. This can vary from 10 to 15% for small wood gasifier-diesel engine generation sets to 30 to 35% for integrated large steam turbine systems. To estimate the quantity of residues needed to supply a desired amount of electricity Figure II-8 can be used. In this Figure, the required gross heat generation can be determined for a desired electrical generation rate. This gross heat generation can in turn be related to residue production rate using Figure II-7.

Thermal energy is generated in addition to electrical energy because of unavoidable inefficiencies of a conversion system. The ratio of electrical to thermal energy is strongly dependent on both the cogeneration cycle that is used as well as the scale and configuration of the wood conversion system. A study by Resource Planning Associates for fossil fuel fired cogeneration systems showed that topping cycles, in which process steam is produced down stream of the electricity generation unit, produced an electrical to thermal energy ratio of 0.25 for steam turbines, 0.6 for gas turbines and 1.4 for diesel engine electricity generation units. This ratio strongly depends on the sophistication of the cogeneration system and the efforts made to recover low-grade heat for process use, or space heating, etc.

The rationale behind selection of an electricity generating system that also matches, or nearly matches, thermal energy requirements is discussed more fully in Chapter III.

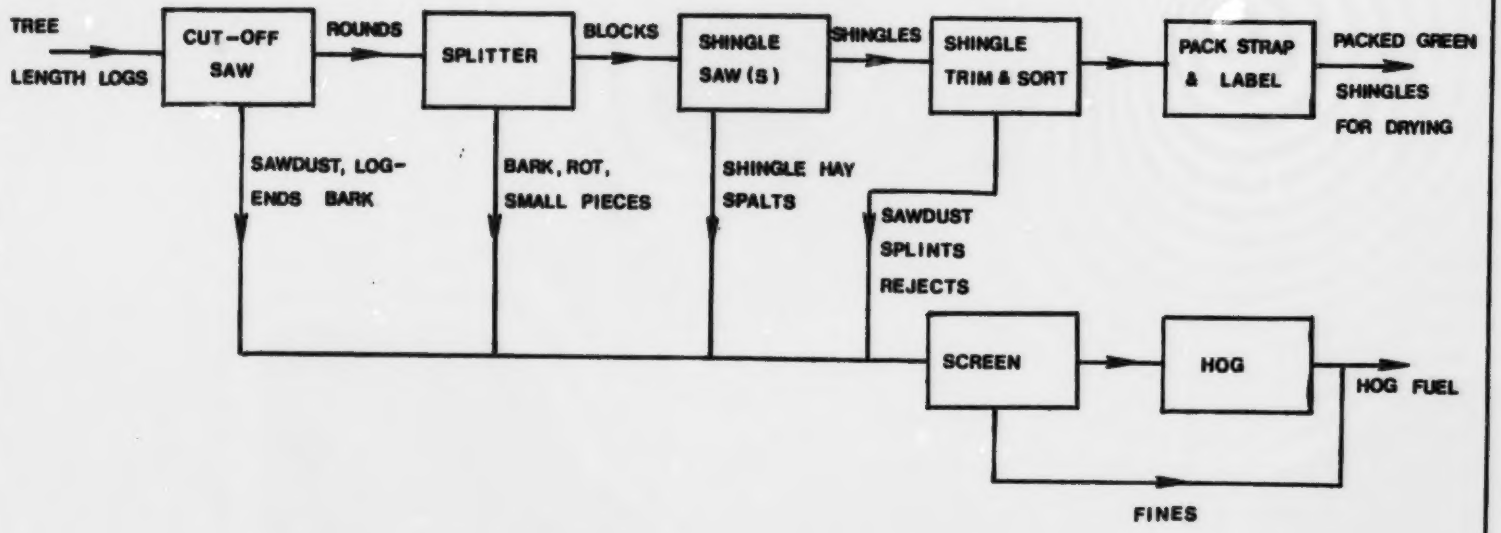


FIGURE II-6 FLOWSHEET OF TYPICAL WESTERN RED CEDAR SHINGLE MILL PRODUCING SHINGLES AND HOG FUEL

Table II-10

PRODUCT YIELDS FOR RED CEDAR SHINGLE MILLS

ITEM	PRODUCT YIELD OD kg/OD kg SHINGLES (ODT/SQUARE)
Reference	Dobie and Wright
Location	British Columbia
Residues:	
Sawdust	0.816 (0.050)*
Knee bolter & Splitter	0.102 (0.006)
Spalts	0.219 (0.013)
Splints	0.351 (0.022)
Total	1.488 (0.091)
Shingles	1 (0.061)

* Shingle type: Fivex @ 0.183 m^3/square and 304 kg/m^3

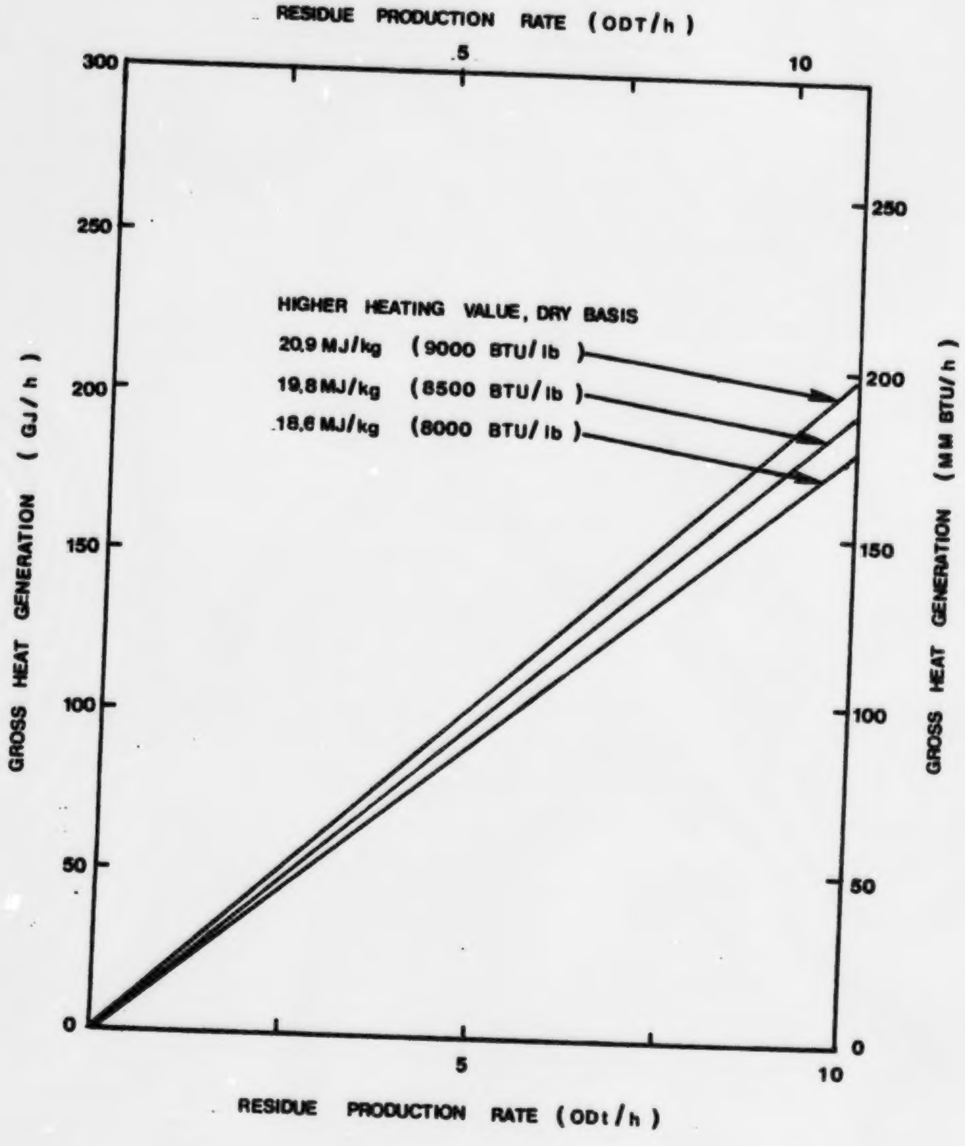


FIGURE II-7 GROSS HEAT GENERATION FROM RESIDUES

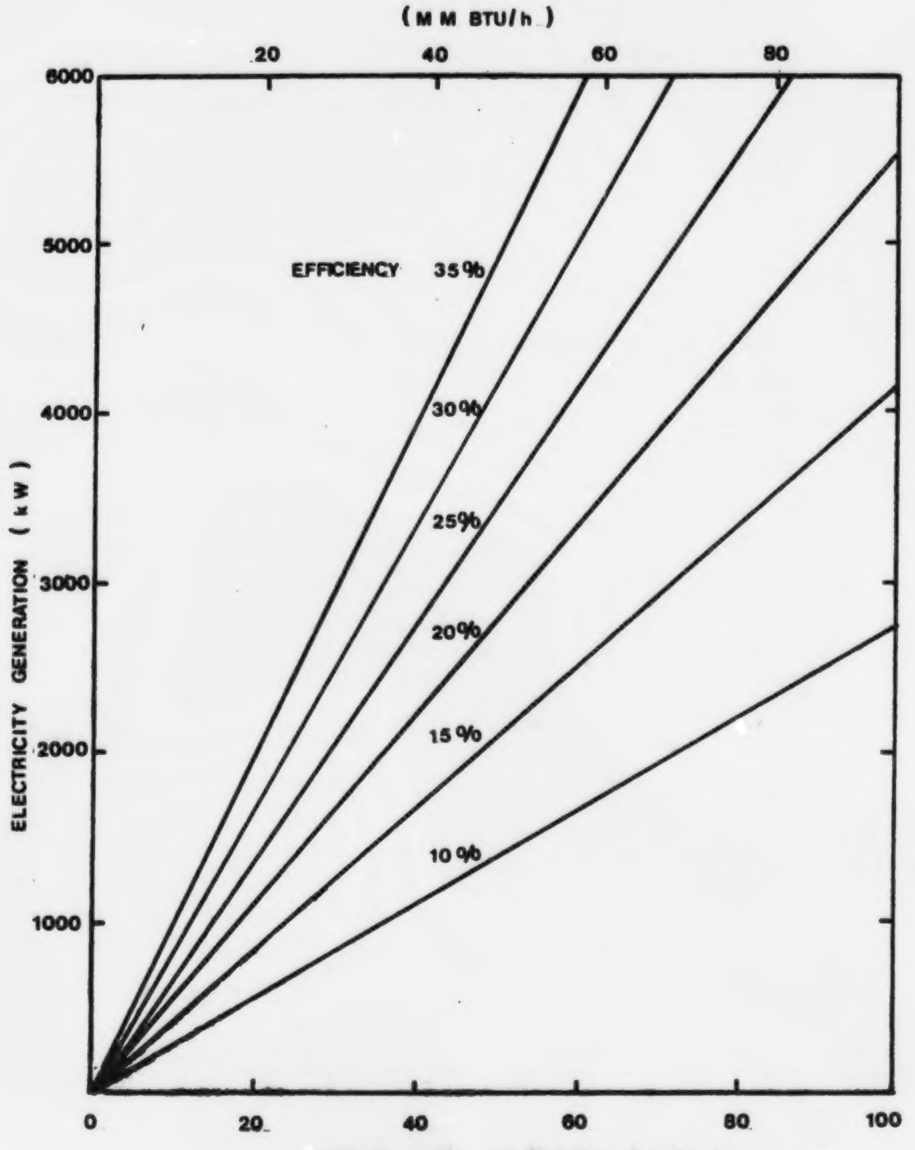


FIGURE II-8 CONVERSION OF THERMAL ENERGY TO ELECTRICITY

C. Competing Sources of Electricity

Several alternative sources exist for industrial electric power, these being self-generation using commercially proven diesel, propane or natural-gas fueled generator sets, and purchase of power from a provincial utility company or some similar large supplier.

Current costs of electricity from provincial utilities in Canada are summarized in Table II-11. Also tabulated is the existing policy of the power company for the purchase of power from an industrial power-generating system connected for return of power to the distribution power grid.

Most of the provincial utilities expressed interest in having electrical generation plants feed power back to the supply grid, with a preferance towards plants that could guarantee delivery at a negotiated capacity. Logically, all utilities are interested in purchase of electricity at a rate below their incremental generation cost. Provinces having established prices for purchase of power are: British Columbia, Alberta, Ontario, and New Brunswick. The other provinces would be willing to negotiate with a potential electricity supplier to attempt to obtain a mutually agreeable price. The provinces not having existing policies had not had sufficient requests for connection of industrial-generating plants to necessitate the development of such a policy.

The purchase prices for power shown in Table II-11 are not firm prices, but ones that would be expected for a typical system. Of particular interest is the comparatively low price offered by Alberta Power Ltd., but it must be emphasized that all the tabulated prices must be used with discretion since substantial changes are possible upon direct negotiation with the power company about a planned generating system.

Power costs for two sizes of sawmills were determined for all provinces and the Yukon Territories. The cases considered were mills of 60,000 m^3/yr (25 MM bf/yr) and 240,000 m^3/yr (100 MM bf/yr) capacity. Billing demand and energy use are typical of these sizes of mill. The calculated power costs use the rates given in Table II-11 and do not incorporate any discounts or special service charges. Discounts are given in all provinces if the mill owns and maintains its own tranformers. Power costs (Table II-12) were lower for the larger of the two mills and varied substantially from province to province. Highest power costs were in the provinces where generation is not hydrobased, but by fossil fuels.

Any self-generated power must compete favourably with the level of costs indicated in Table II-12 or offer other tangible benefits, such as improved service, or isolation from escalating fuel prices, or reduction of wood waste handling and disposal.

Table II-11

COSTS OF ELECTRICITY IN CANADA BY PROVINCE

ITEM	PROVINCE & COMPANY	BRITISH COLUMBIA B.C. HYDRO & POWER AUTHORITY		ALBERTA ALBERTA POWER	
Effective Date		February 1981	April 1981	October 1981	October 1981
Reference		1200, 1201, 1210, 1211	1821	31	32
Load Range		>35 kW	>5000kW	Main Transmission	Subtransmission
Electricity Cost (per mo)					
Demand Charge		$3/mo plus 35 - 150 kW @ $2.40/kW >150 kW @ $4.60/kW	$3/kVA	0 - 300 kW @ $3,873 300 - 1000 kW @ $7/kW >1000 kW @ $2.69/kW	0 - 300 kW @ $3,970 300 - 1000 kW @ $7.27/kW >1000 kW @ $2.97/kW
Energy Charge		0 - 275 kWh @ 5.389¢/kW 275 - 7000 kWh @ 3.889¢/kW 7000 - 30,000 kWh @ 2.789¢/kW >30,000 kWh @ 1.689¢/kW	1.38¢/kWh	0 - 400 kWh/kW @ 2.8¢/kWh >400 kWh/kW @ 1.55¢/kWh	0 - 400 kWh/kW @ 2.91¢/kWh >400 kWh/kW @ 1.66¢/kWh
Definitions			Demand is max. of: max. monthly demand; 75% max. demand in prior 12 mo; 5000 kVA	Demand is max. of: max. monthly demand; max. demand in prior 12 mo excluding April - Sept.; 300 kW	Demand is max. of: max. monthly demand; max. demand in prior 12 mo excluding April - Sept.; 300 kW
Policy For Purchase of Power By Utility		Requests assessed individually. Under continual review	Requests assessed individually. Under continual review	Requests assessed individually.	Requests assessed individually.
Power Purchase Price (¢/kWh)		1.4 ¢/kWh	1.4 ¢/kWh	0.47 ¢/kWh	0.47 ¢/kWh

Table II-11 Continued

COSTS OF ELECTRICITY IN CANADA BY PROVINCE

ITEM / PROVINCE & COMPANY	ALBERTA TRANS ALTA UTILITIES	SASKATCHEWAN SASKATCHEWAN POWER CORP.		MANITOBA MANITOBA HYDRO
Effective Date	October 1981	1981	1981	1981
Reference	790	4	5	PD148A
Load Range	>50 kW	75 - 1000 kVA	1000 - 7500 kVA	>50 kVA
Electricity Cost (per mo)				
Demand Charge	0 - 500 kW @ $10.65/kW 500 - 2500 kW @ $8.35/kW >2500 kW @ $5.90/kW	$5.85/kVA	$5.70/kVA	Standard: 0 - 50 kVA $8.35+; >50 kVA $4.75/kVA Primary: $8.35+; $4.30/kVA
Energy Charge	0 - 400 kWh @ 1.1¢/kWh >400 kWh @ 0.6¢ /kW	1.7¢/kWh	5.6¢/kWh	0 - 1200 kWh @ 4.7¢/kWh 1200 - 11,700 kWh @ 2.97¢/kWh >11,700 kWh @ 1.124¢/kWh
Definitions	Demand is max. of: max. monthly demand; 85% max. demand in prior 12 mo; 50 kW	Demand is max. of: max. monthly demand; 75 kVA	Demand is max. of: max. montly demand; 50% max. demand in prior 12 mo	Demand is max. of: max monthly demand; max. 80% of max. demand in prior winter months; 25% contract demand
Policy For Purchase of Power By Utility	Requests assessed individually.	No existing policy		No existing policy
Power Purchase Price (¢/kWh)	By contract	---		---

Table II-11 Continued

COSTS OF ELECTRICITY IN CANADA BY PROVINCE

ITEM	PROVINCE & COMPANY	ONTARIO ONTARIO HYDRO	QUEBEC HYDRO-QUEBEC	NEW BRUNSWICK N.B. ELECTRIC POWER COMM.	PRINCE EDWARD ISLAND MARITIME ELECTRIC CO. LTD.
Effective Date		January 1981	January 1982	May 1981	October 1981
Reference			Bylaw 290		
Load Range		<5,000 kW	100 - 5,000 kW	>750 kW	>750 kW
Electricity Cost (per mo) Demand Charge		$4.00/kW	$3.21/kW	0 - 2500 kW @ $6.85/kW 2500 - 7500 kW @ $6.67/kW >7500 kW @ $6.24/kW	$2.97/kWA
Energy Charge		0 - 250 kWh @ 11.4¢/kWh 250 - 10,000 kWh @ 3.95¢/kWh 10,000 - 2,000,000 kwh @ 1.95¢/kWh	0 - 120 hr of billing demand @ 4.3¢/kWh net 87,000 kWh @ 2.6¢/kWh remainder @ 1.82¢/kWh	0 - 900 MWh @ 2.53¢/kWh 900 - 2700 MWh @ 2.41¢/kWh >2700 MWh @ 2.16¢/kWh	0 - 200 kWh/kW @ 8.923¢/kWh >200 kWh/kW @ 7.605¢/kWh
Definitions			Demand is max. of: max. monthly demand; contract demand	Demand is max. of: max. monthly demand; 90% maximum kVA; 90% of contracted kW	Demand is max. of: max. monthly demand; 80% max. demand in prior12 mo; 600 kVA
Policy For Purchase of Power By Utility		Established policy for purchase.	No established policy. Requests assessed individually	Established policy for purchase.	No existing policy. Requests assessed individually.
Power Purchase Price (¢/kWh)		Winter -$3.34/kW + Peak - 1.74¢/kWh Off-Peak - 1.66¢/kWh Summer - $1.60/kW + Peak - 1.534¢/kWh Off-Peak - 1.34¢/kWh		Changes monthly. Typical 1981 price 1.803¢ - 2.718¢/kWh Approx. 70% of incremental generation cost.	Expected to be near incremental generation cost.

Table II-11 Continued

COSTS OF ELECTRICITY IN CANADA BY PROVINCE

ITEM	PROVINCE & COMPANY	NOVA SCOTIA N.S. POWER CORP.	NEWFOUNDLAND NFLD & LABRADOR HYDRO	YUKON TERRITORIES YUKON ELECTRIC CO.	NORTH WEST TERRITORIES NORTHERN CANADA POWER COMM.
Effective Date		June 1981	January 1981	June 1981	April 1981
Reference		22	Industrial Rate 1	Whitehorse Gen. Ser. Y-21	Pine Point Mines
Load Range		250 - 3999 kVA	<10,000 kW		
Electricity Cost (per mo)					
Demand Charge		$4.87/kVA	$3.75/kVA	$1/kW	$14.95/kW
Energy Charge		2.75¢/kWh	0 - 150 kWh/kW @ 4.2¢/kWh >150 kWh/kW or 90,000 kWh @ 2.2¢/kWh 0.5 - 0.7¢/kWh fuel surcharge	0 - 200 kWh/kW @ 6.66¢/kWh 200 - 400 kWh/kW @ 5.67¢/kWh >400 kWh/kW @ 4.57¢/kWh 0.596¢/kWh fuel surcharge	Hydro Energy @ 1.36¢/kWh Diesel Energy @ 7.86¢/kWh (variable)
Definitions		Demand is max. of: max. monthly demand; max. demand in Dec., Jan., Feb. or March.	Demand is max. of: max. demand in prior 12 mo.	Demand is max. of: max. demand; max. demand in prior 12 mo; 5 kW	Demand is max. of: max. demand; max. demand in prior 12 mo.
Policy For Purchase of Power By Utility		No existing policy.	Requests assessed individually.	Requests assessed individually.	Requests assessed individually.
Power Purchase Price (¢/kWh)		---	Near incremental generation cost.	By negotiation.	Lower than incremental generation cost.

Table II-12

TYPICAL COSTS FOR POWER FROM UTILITY COMPANIES FOR TWO SIZES OF SAWMILLS

	CASE 1	CASE 2
Mill Production (m^3/yr)	60,000	240,000
Billing Demand (kW)	1,300	4,500
Energy Use (kWh/month)	200,000	1,500,000
Overall Power Cost ($/kWh)		
British Columbia (B.C. Hydro)	0.0456	0.0305
Alberta (Alberta Power)	0.0789	0.0421
Saskatchewan (Sask. Power)	0.0931	0.0731
Manitoba (Manitoba Hydro)	0.0422	0.0255
Ontario (Ontario Hydro)	0.0456	0.0315
Quebec (Hydro-Quebec)	0.060	0.0418
New Brunswick (N.B. Power)	0.0698	0.0452
P.E.I. (Maritime Electric Co.)	0.1085	0.0929
Nova Scotia (N.S. Power Corp.)	0.0591	0.03889
Newfoundland (NFLD Power & Labrador Hydro)	0.0614	0.0405
Yukon Territories (Yukon Electric Co.)	0.0791	0.0716

It may be necessary to interface a mill's generating facility with a provincial utility to avoid power outages. This requires appropriate equipment to protect both power sources. Sale of power from the mill's power system requires additional sophistication of this interface to measure the amount of power delivered. Ontario Hydro has published a statement of the technical requirements for interfacing a mill-based power plant with another power source, such as a provincial utility. For small systems the cost of this interface is not great. These requirements are given in Table II-13.

In addition to the capital expense of items needed for interfacing the parallel generators, the mill would be expected to cover the costs of equipment needed by the utility to purchase power from the mill as well as equipment needed by the utility for protection and safe operation. Cost sharing arrangements would have to be negotiated individually and will likely differ between provinces as well as between mills.

Table II-13

TECHNICAL REQUIREMENTS FOR PARALLEL GENERATION OF ELECTRICITY

1. A means of synchronizing the generator and the power system;
2. A circuit breaker between the generator and the power system, with adequate interrupting capacity to clear all faults on either side of the breaker;
3. Generator overload protection;
4. Provision for automatic disconnection of the generator from the power system in the event of an outage of the power system supply;
5. Provision for isolation of the generator from the power system in the event of unsatisfactory generator voltage or frequency;
6. A visible, manual disconnect switch close to the point of interconnection of the generator and the power system. This provides utility maintenance crews a means of isolating the generator when work on power system equipment is necessary;
7. All equipment must be CSA approved and installed in accordance with the Electricity Safety Code;
8. All equipment must be designed to avoid damage from over-voltage or automatic reclosure operations on the power system;
9. The harmonic content of the generator output must be within acceptable limits.

Source: Ontario Hydro (March, 1981)

III ALTERNATIVES FOR GENERATION OF ELECTRICITY USING WOOD

A. Concepts for Energy and Material Flows in Mills

Conceptually, a wood industry process operates with two inputs, raw forest material and energy, and three outputs, the final wood product, wood residues, and residual energy. The energy stream into the mill can be separated into thermal energy (process heat and steam and space heating) and electrical energy which is usually purchased from the local electrical utility company. The residual energy output is the waste heat released to the environment. The third output, wood residues, are used as landfill, sold as hog fuel, or as a by-product for industrial use or used as a fuel for the on-site generation of electrical or thermal energy. Figure III-1 shows a typical industrial process having no waste heat recovery or wood residue utilization.

The basic philosophy of cogeneration of electrical and thermal energy involves recovering the residual energy in either or both of two techniques known as topping and bottoming cycles. In a bottoming cycle (Figure III-2) wood residue may be used to fuel a thermal energy generator, for example a waste wood boiler. This thermal energy is then used for some process requirement from which waste heat is released. This waste heat may then be converted to electrical energy using a suitable prime mover, for example a steam turbine. It should be noted that in the forest industry, this waste heat is generally of very low temperature, and its recovery using a bottoming cycle is not likely.

In a topping cycle (Figure III-3) wood residue is used to fuel the electrical energy generator, such as an indirectly fired gas turbine. The thermal energy generated from the system's prime mover supplies the steam or heat load required for the process.

Electrical and thermal energy cogeneration using a combined topping and bottoming cycle is illustrated in Figure III-4. In this case wood residues produced from the industrial process fuels the topping electrical energy generation system. Excess thermal energy produced by this system not used for the process is then converted into electrical energy in the bottoming cycle.

B. Systems for Electricity Generation

1. System Options

When proven and unproven technology is included, there are many options for cogenerating electricity and heat from wood residues. Because of this selection, the characterization and evaluation of the systems was done using a unit process approach which facilitates the assembly of any system that may be desired.

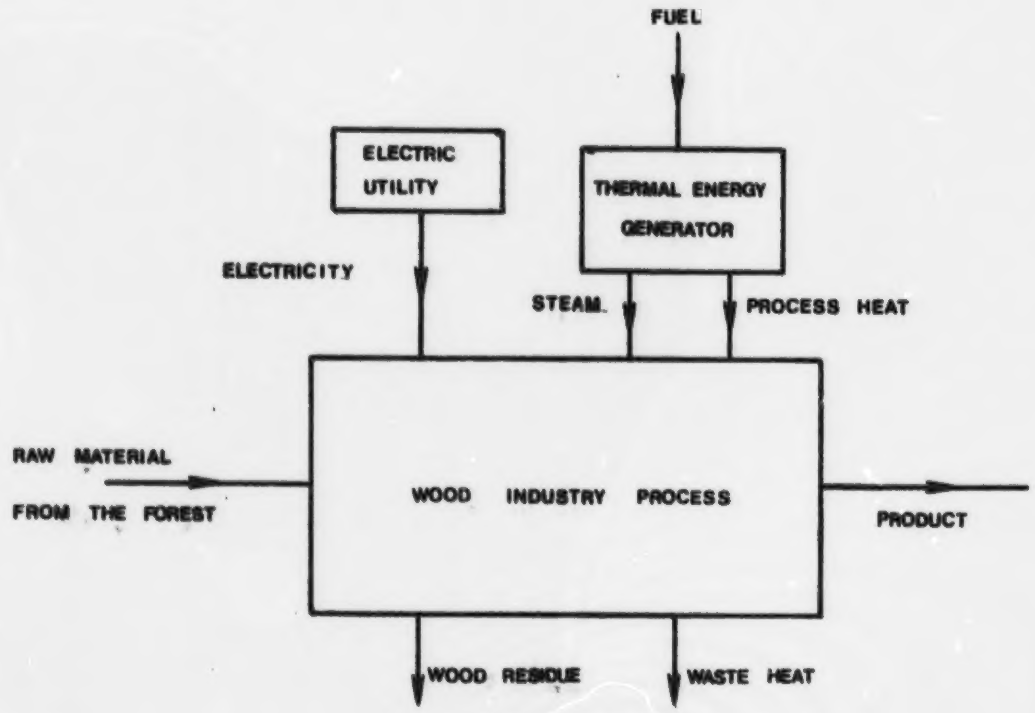


FIGURE III-1 . CONVENTIONAL PROCESS WITHOUT RESIDUAL ENERGY RECOVERY

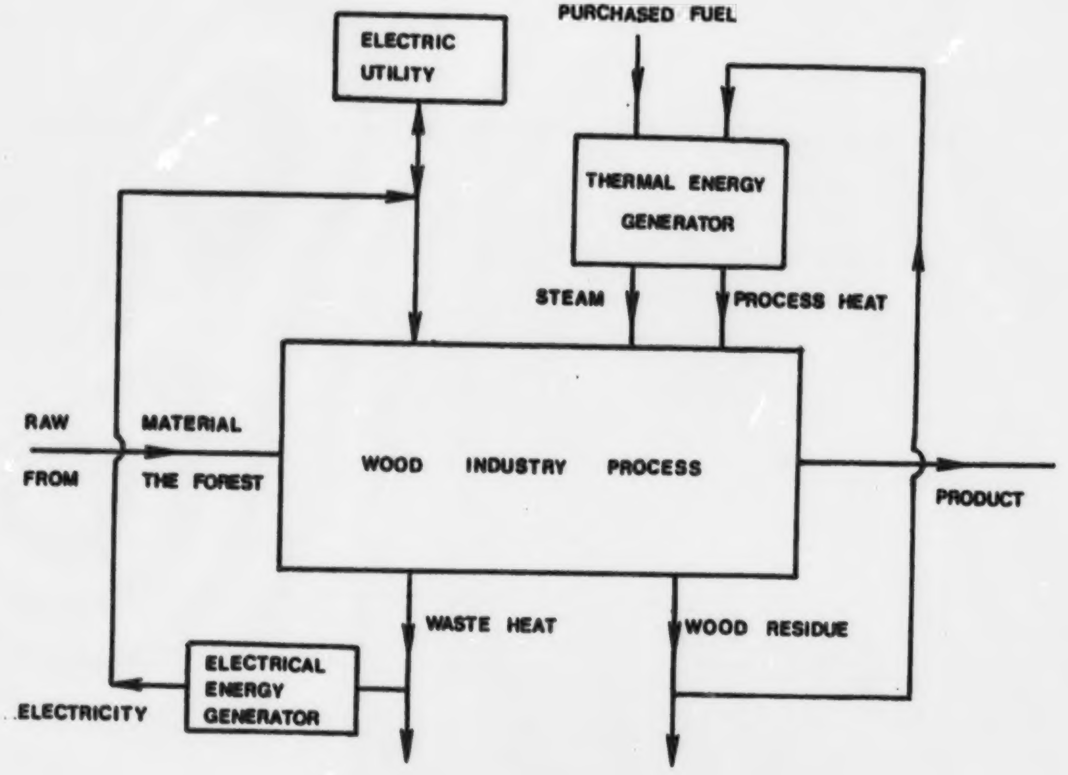


FIGURE III-2 CO-GENERATION USING A BOTTOMING CYCLE

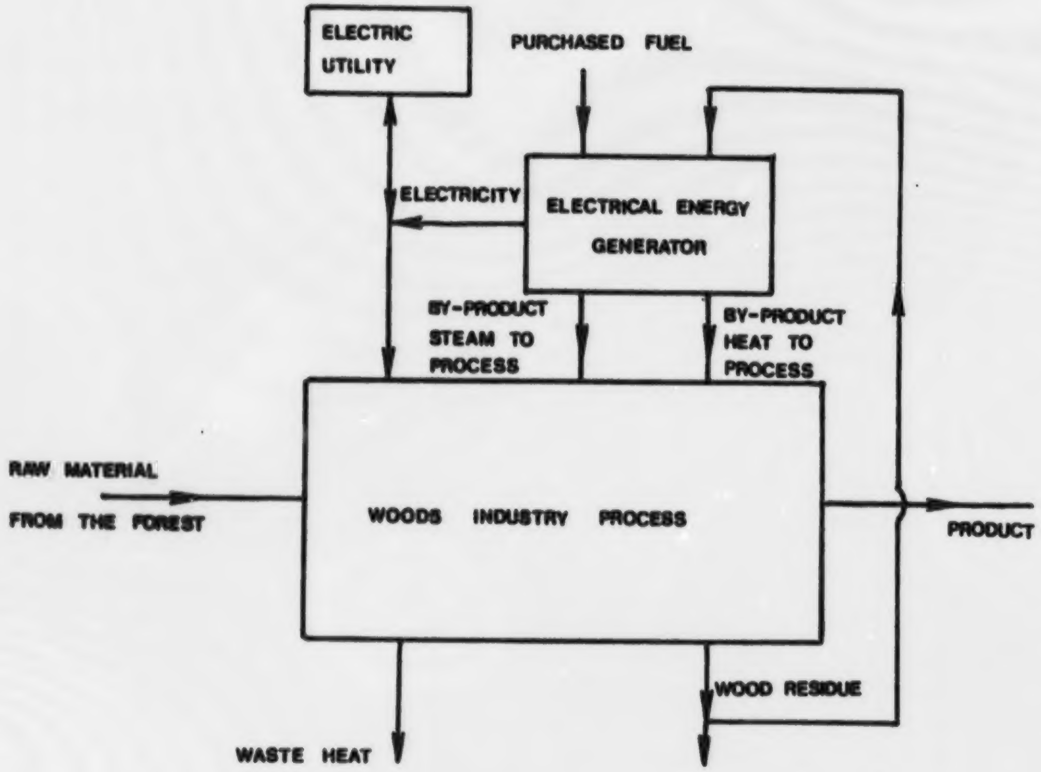

ELECTRIC UTILITY
PURCHASED FUEL
ELECTRICITY
ELECTRICAL ENERGY GENERATOR
BY-PRODUCT STEAM TO PROCESS
BY-PRODUCT HEAT TO PROCESS
RAW MATERIAL FROM THE FOREST
WOODS INDUSTRY PROCESS
PRODUCT
WOOD RESIDUE
WASTE HEAT

FIGURE III-3 CO-GENERATION USING A TOPPING CYCLE

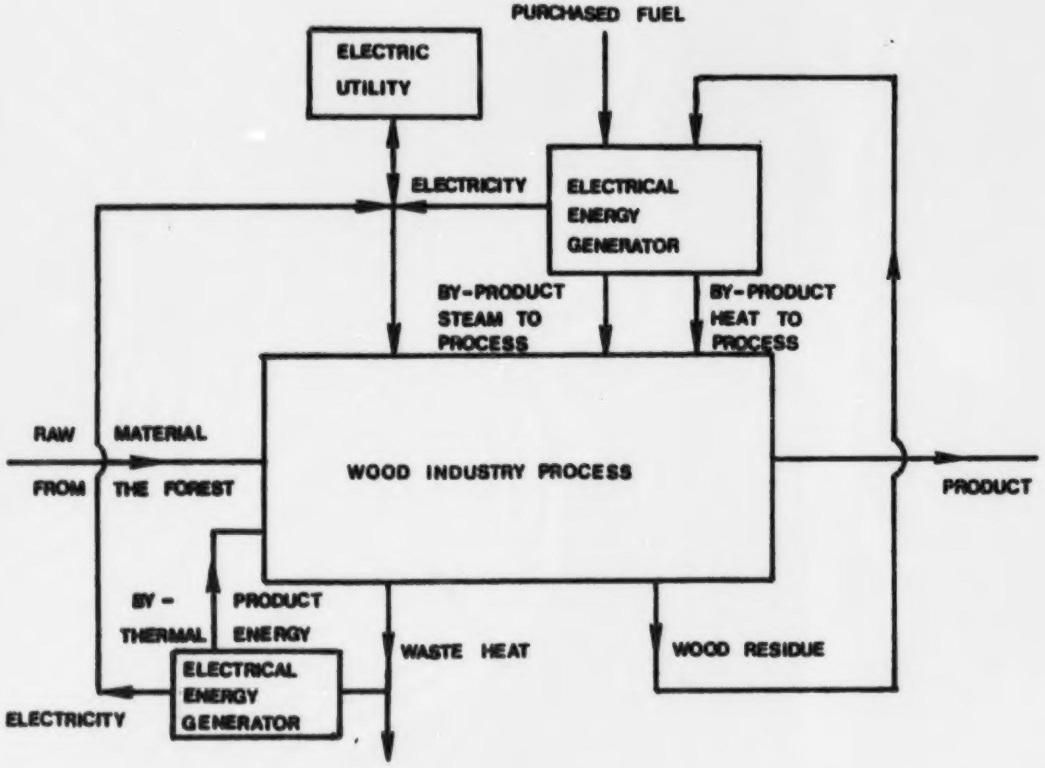

PURCHASED FUEL
ELECTRIC UTILITY
ELECTRICITY
ELECTRICAL ENERGY GENERATOR
BY-PRODUCT STEAM TO PROCESS
BY-PRODUCT HEAT TO PROCESS
RAW MATERIAL FROM THE FOREST
WOOD INDUSTRY PROCESS
PRODUCT
BY-PRODUCT THERMAL ENERGY
WASTE HEAT
WOOD RESIDUE
ELECTRICAL ENERGY GENERATOR
ELECTRICITY

FIGURE III-4 CO-GENERATION USING BOTTOMING & TOPPING CYCLES

Three unit processes were required for the system analysis: Primary Conversion, Secondary Conversion, and Prime Mover. The Primary Conversion process receives wood, and through some means of conversion produces energy in a primary working fluid or energy form; examples are: steam, fuel gas, solid fuel, liquid fuel, and hot gas. This working fluid may be used directly in a prime mover for generating electricity or may be treated by a secondary conversion process to produce a secondary working fluid.

The unit process approach was applied to both topping and bottoming cycles, leading to the summary of system configurations illustrated in Figure III-5.

The most common system for generating electricity from wood is the one utilizing a steam boiler and steam turbine. Systems using a gasifier and internal combustion engine, either spark ignition or diesel cycle, are being demonstrated at several locations in North America, as well as elsewhere, and are an attractive alternative to the conventional steam cycle. The other systems that have been recently proposed for forest products installations are: 1) a gasifier and gas turbine combination; and 2) a combustor, heat exchanger, and gas turbine combination for which air is the secondary working fluid. Most of the remaining configurations have been or are being studied to determine their future potential and, therefore, fall in the class of non-proven and emerging technology.

The following discussion presents system configurations and general performance characteristics of the identified electricity generating options, categorized by type of primary conversion system. Detailed discussions about each of the unit processes and manufacturers' designs for these unit processes are given in succeeding chapters.

2. Steam Boilers

These conversion units as considered here are for application in topping cycles where electricity is usually cogenerated with process steam. Steam boilers used for heat recovery from exhaust gases are treated in the subsection on Bottoming Heat Recovery Boilers.

a) Cycle Configurations

Boilers may be coupled with either steam turbines or steam engines to generate electricity, both prime movers being operated according to a Rankine power cycle. Both prime movers are availale with similar options to allow considerable latitude for integration into a cogeneration system.

Turbines are used exclusively in North America for power generation with steam, this being a logical development

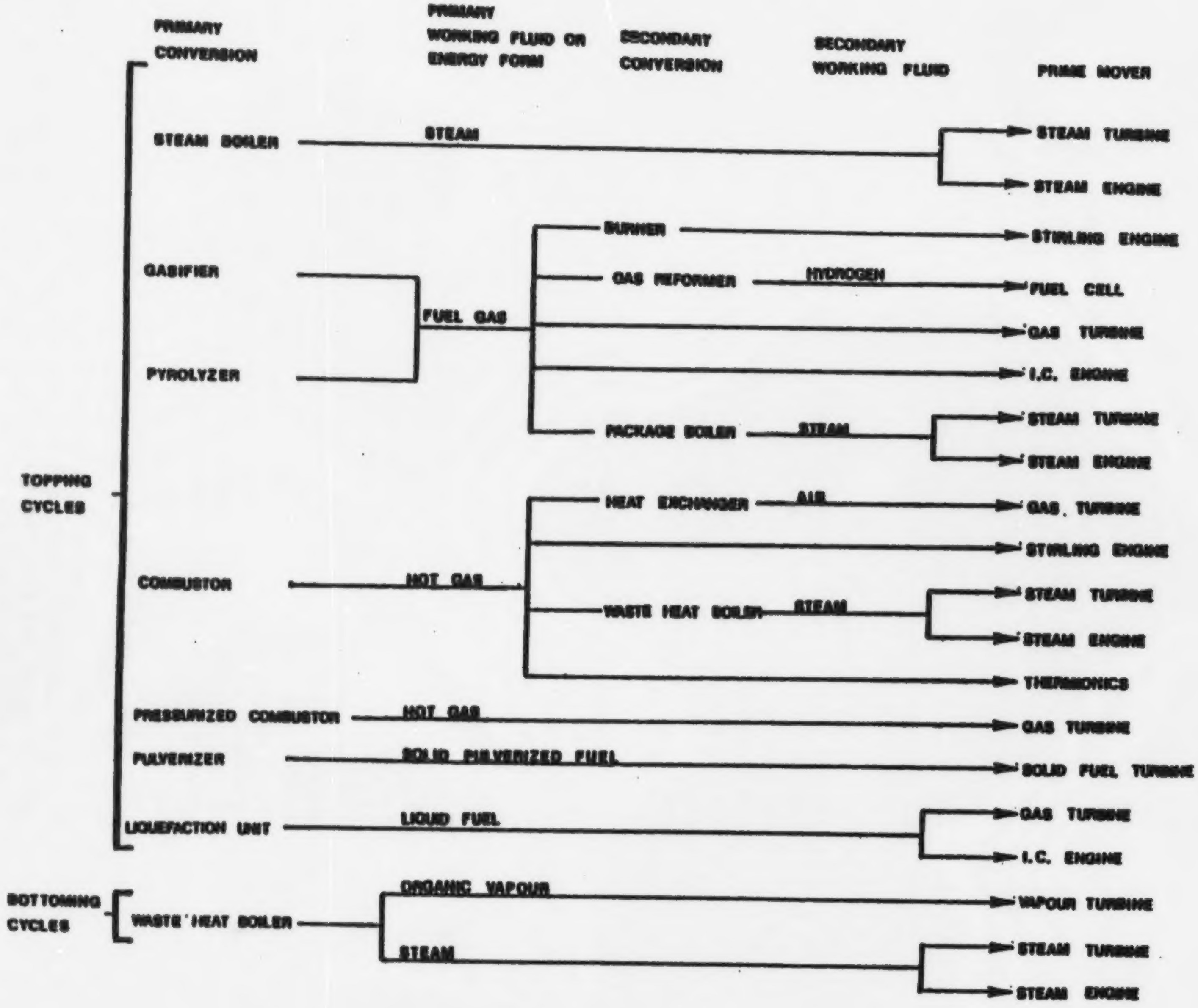


FIGURE III-5 ALTERNATIVE SYSTEM CONFIGURATIONS

because of the advantages of turbines over reciprocating engines. Although many companies manufacture steam turbines, only two companies were located that manufacture steam engines.

The complexity of Rankine power generating cycles increases with output capacity. Systems must be tailored to satisfy both turbine throttle conditions and pressures for process steam, and these requirements make most installations somewhat unique. Systems with less than 5 MW capacity tend to be fairly simple in design for reasons of economy, and do not incorporate cycle features to maximize thermal efficiency such as combined high and low pressure turbines and regeneration using feedwater heaters.

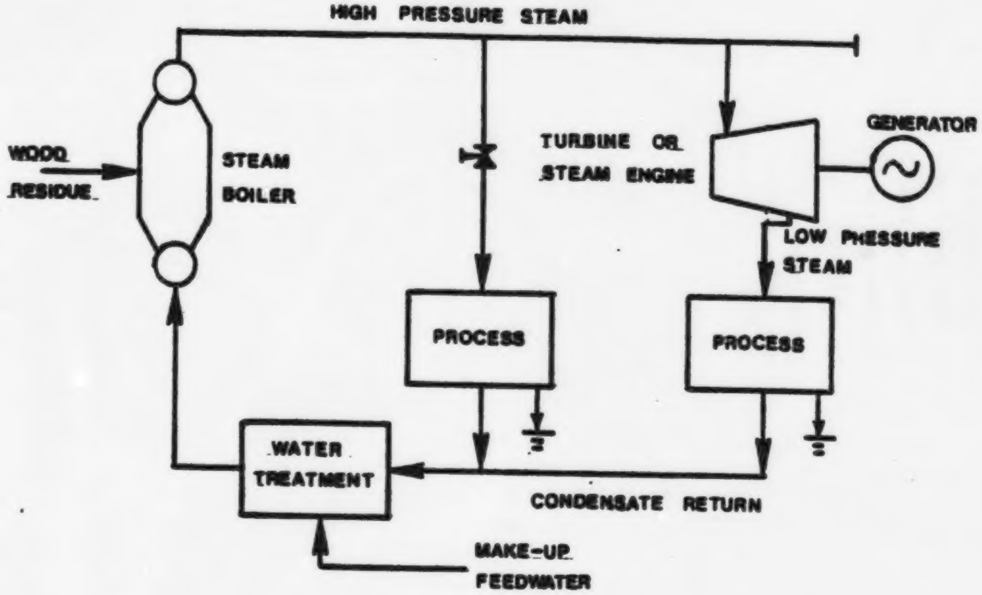
Steam Rankine cycles are usually categorized according to whether a back-pressure or a condensing turbine is used. Back-pressure (non-condensing) turbines exhaust at or above atmospheric pressure. Condensing turbines exhaust steam at sub-atmospheric pressure and may allow extraction of steam at a desired pressure from some location in the turbine expansion section. Basic configurations of a steam cycle with a back-pressure turbine and a condensing/extraction turbine are shown in Figure III-6.

Condensing/extraction turbines generally cost near twice that of straight-through flow back-pressure or condensing multiple-stage turbines. It may be more economical to use two turbines, one of back-pressure design and one of condensing design, arranged in parallel, rather than a condensing/extraction turbine.

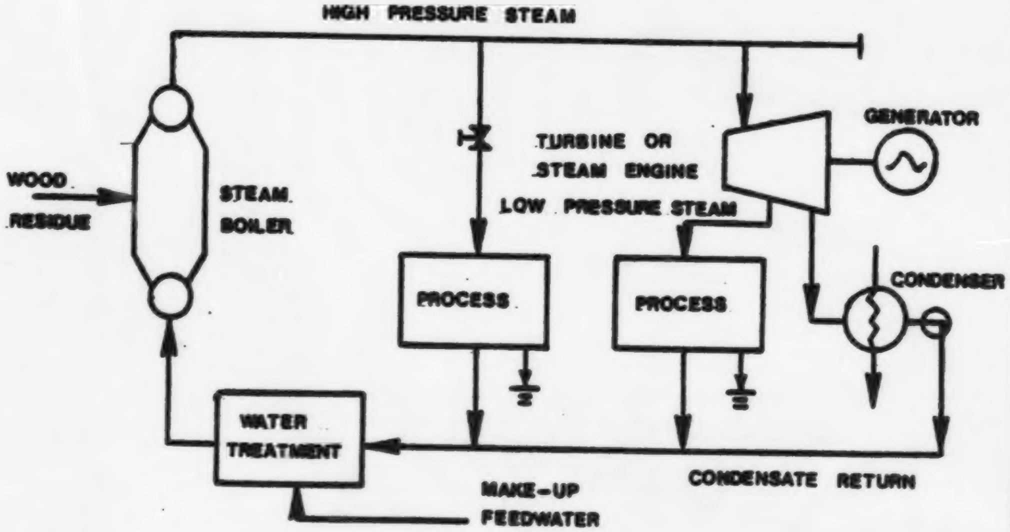
Steam engine power generating systems are fundamentally identical to turbine systems, and the following discussion of turbine performance generally applies to steam engines. Figure III-6 presents simple system designs for back-pressure and condensing engines. For outputs less than 2 MW, steam engines should be carefully considered as an alternative to a traditional turbine. Their main advantages over turbines are excellent torque characteristics, especially at low speed, and operation with variable speeds. These advantages make engines certainly attractive as mechanical drives, but are of little benefit for constant speed operation with a generator.

b) General System Characteristics

The relationship between steam flow and output power for a condensing turbine and two back-pressure turbines, one exhausting at 345 kPa, the other at 1,035 kPa are given in Figure III-7 for single and multiple-stage designs. These



A. BACK PRESSURE TURBINE



B. CONDENSING / EXTRACTION TURBINF

FIGURE III-6 STEAM TURBINE TOPPING RANKINE CYCLES

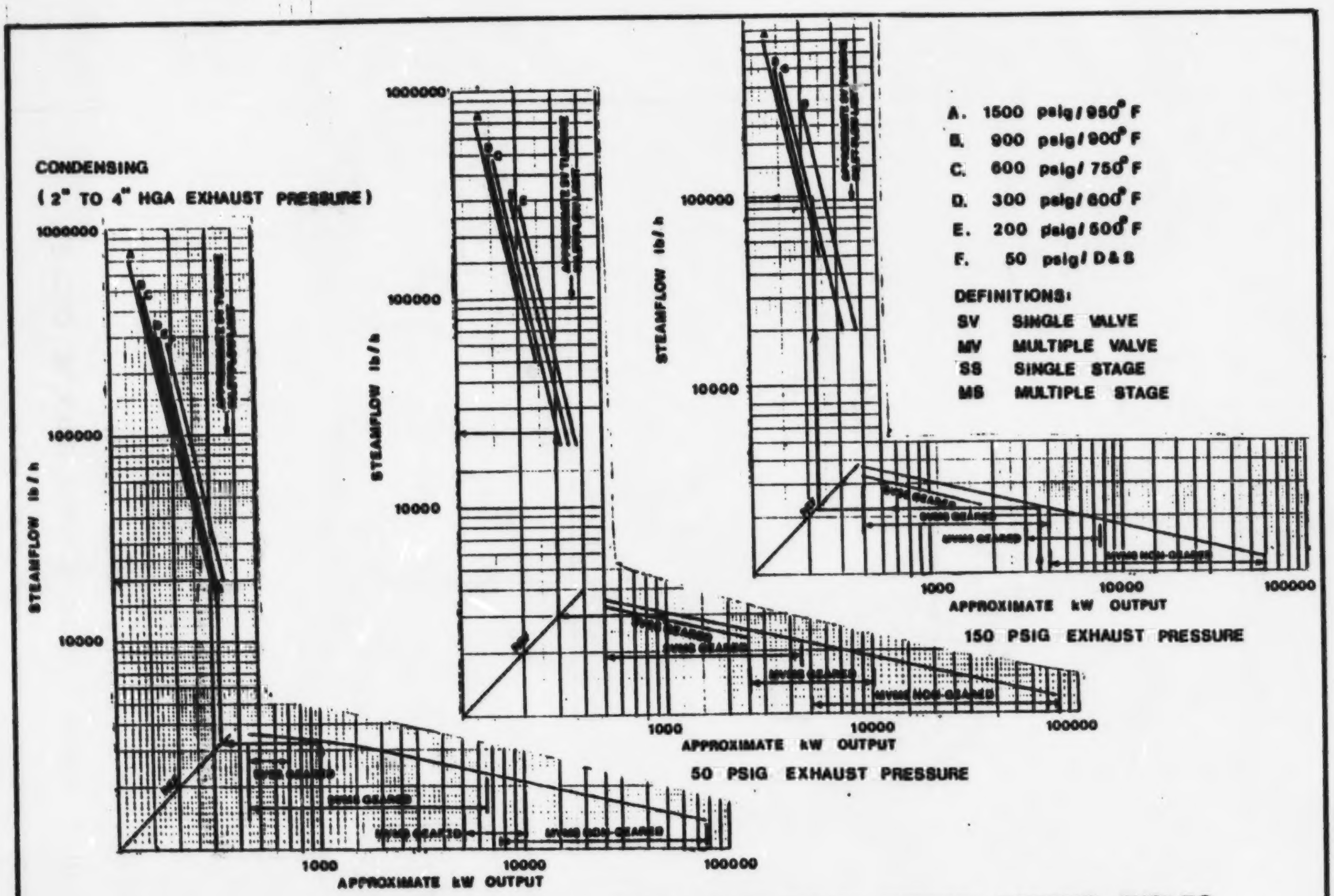


FIGURE III-7 APPROXIMATE STEAM RATES FOR STEAM TURBINE TOPPING CYCLES

SOURCE: BULLETIN H-43, UNITED TECHNOLOGIES, ELLIOTT CO.

estimating curves allow a quick illustration of the difference in steam requirements of the condensing designs versus the non-condensing designs. For example, a 1 MW condensing multiple-stage turbine requires a steam flow of 9,000 kg/h, compared to 11,000 kg/h for a 345 kPa back-pressure unit, and 13,600 kg/h for a 1,035 kPa back-pressure unit, of the same power output capacity. This difference in steam rate is often the deciding factor in selecting the turbine design compatable with a mill.

The ratios of electrical to thermal energy for condensing and back-pressure turbine designs that are shown in Figure III-8 for common inlet steam conditions exemplify the advantage of the condensing design if process steam requirements are moderate. The data show back-pressure designs to have an E/T of between about 0.04 and 0.2, depending on inlet steam conditions. Condensing turbines have an E/T of 0.25 to 0.43 increasing with increasing pressure. The gap shown between the E/T of condensing and back-pressure turbines is bridged by condensing/extraction turbines. The range of E/T possible with steam turbines makes them well suited to pulp and paper mills, which have an E/T normally near 0.1 to 0.2.

One of the influential factors in selecting the type of steam turbine that best suits an application is the availability of standard units. Condensing turbines are available in sizes from a 300 kW to over 100,000 kW, possibly incorporating single-stage expansion for units below 700 kW. Larger condensing turbines are all of multiple-stage design. The smallest back-pressure turbines are of single-stage design with output capacities of 300 kW to over 2,500 kW. Outputs in this range can also be achieved with multiple-stage units with less steam consumption than single-stage units. Multiple-stage turbines are commonly available in sizes from 1,500 kW up to and exceeding 100 MW, and are preferred over single-stage designs unless the larger steam requirements of the single-stage designs can be effectively used for process applications.

Although considerable flexibility of operating conditions can be achieved with steam turbines by appropriate variations of inlet and outlet and, possibly, extraction conditions, it is important that the installation be designed to satisfy the mill demand for both electricity and process heat/steam. If no steam demand exists, then a condensing turbine is most appropriate, although these units still exhaust large amounts of hot water that must be cooled. This heat is suitable for space heating or similar heating applications.

If process steam is required, either a back-pressure or a condensing/extraction turbine may be chosen. A back-pressure

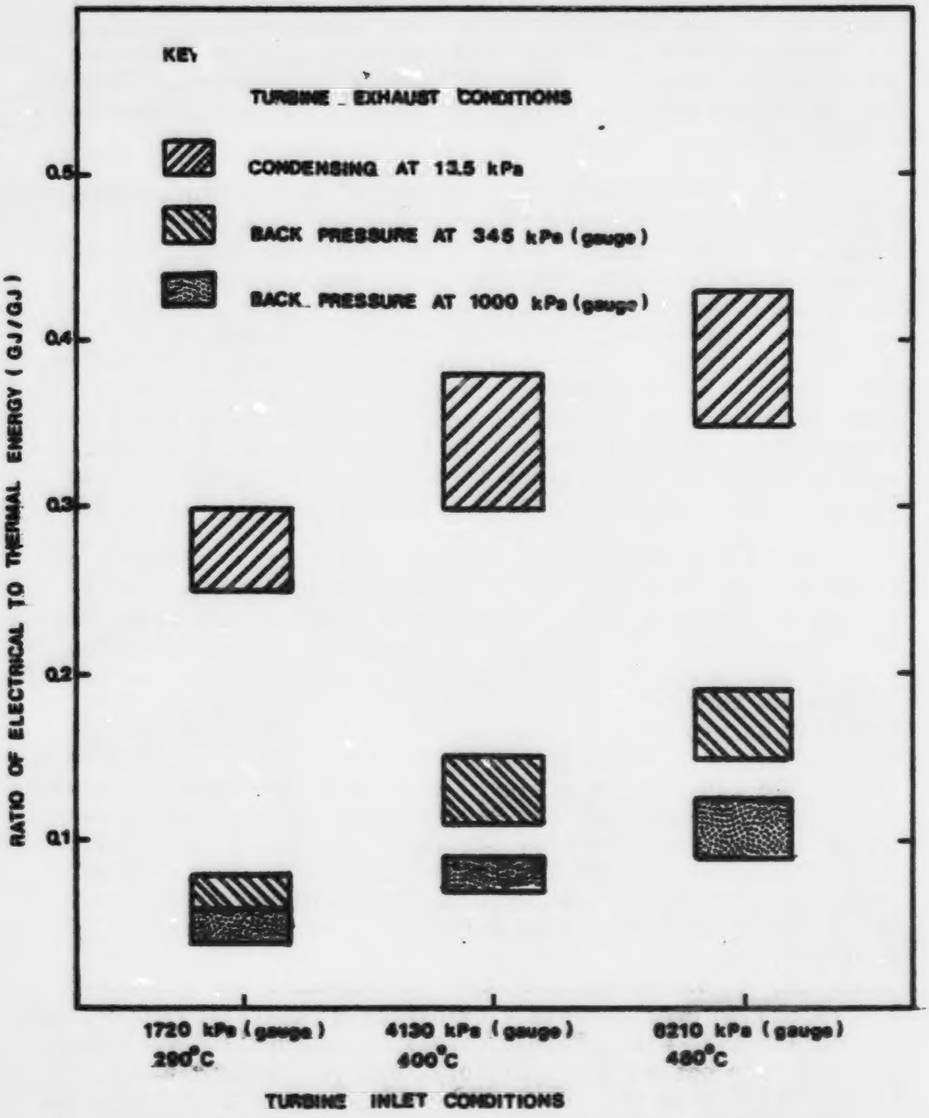


FIGURE III-8 RATIO OF ELECTRICAL TO THERMAL ENERGY FOR STEAM TURBINES

unit operates best with fairly steady steam and electricity loads, and with these loads occurring simultaneously. Outputs of steam and electricity in proportions substantially different than possible with the turbine require special system features to accommodate either excess steam or excess electricity generaton. Condensing/extraction turbines offer the best operating flexibility, but at increased cost.

In an analysis of cogeneration at a forest products mill, Archibald and Gabriel state several factors that improve the economics of such a project:

- high process steam requirements, usually in excess of 45,000 kg/h;
- low process steam pressure;
- a good plant load factor, preferably in excess of 6,000 days per year;
- plant lifetimes of 10 to 20 years;
- good electrical tie-in costs and reliability;
- investment tax credits and write-offs.

Compared to other elctricity generating systems, steam based ones require more highly trained operating personnel and greater supervision as a consequence of the provincial boiler codes. In a study of a 250 kW application with 1,550 kPa, 205°C and 4,500 kg/h fire-tube boiler with back-pressure turbine, SNC put the required number of operating personnel at 3.6 per 100 kW. At an annual salary of $25,000, this requirement amounts to $0.14/kWh for 6,500 operating hours per year. Labour requirements must be calculated for each installation because of differences in the boiler codes by province (see Section C-2). They will not vary proportionally with system capacity. The manpower requirement of a 2,000 kW power plant would be about 1.1 per 100 kW, having an equivalent cost of $0.04/kWh.

Steam engines require greater space than a turbine and have higher thermal losses and, for larger sizes, are more expensive. Steam engines have a lower steam rate than turbines in sizes under about 2 MW. In sizes up to 20 MW, both designs have similar steam rates, while over 20 MW turbines have lower steam rates. Steam engines operate at low speeds, typically below 1,200 rpm, and, consequently, major overhauls are very infrequent. Routine daily maintenance is needed for lubrication, etc.

Because of boiler codes, steam engine systems would need manpower similar in number to turbine systems.

3. Gasifiers and Pyrolyzers

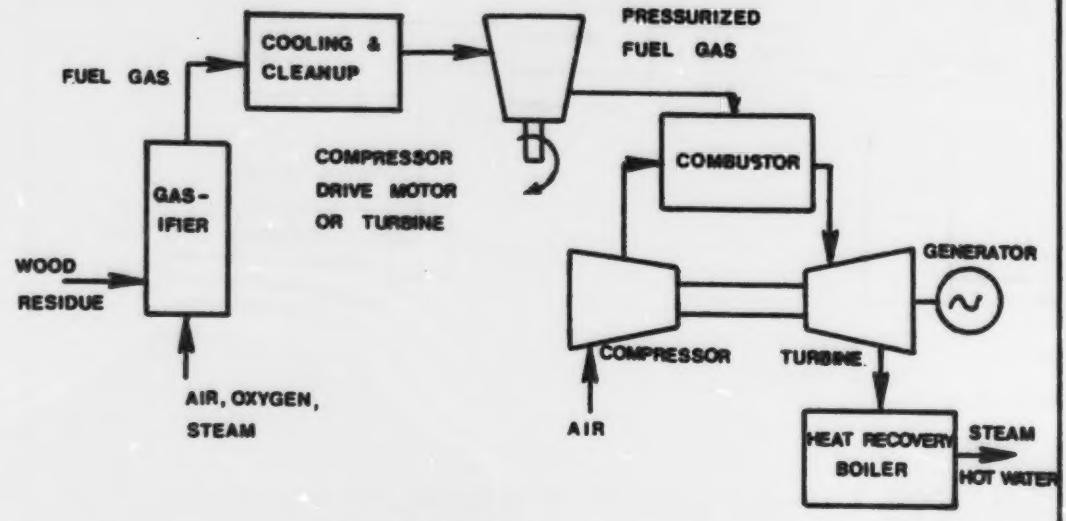
Several systems are possible with the use of fuel gas of a composition produced by gasifiers or pyrolyzers. The discussion here emphasizes gasifiers solely because these systems emphasize production of fuel gas. Gas from pyrolyzers contains more tar than gas from gasifiers, and consequently, more capacity would be needed in a gas cleanup system.

a) Cycle Configurations

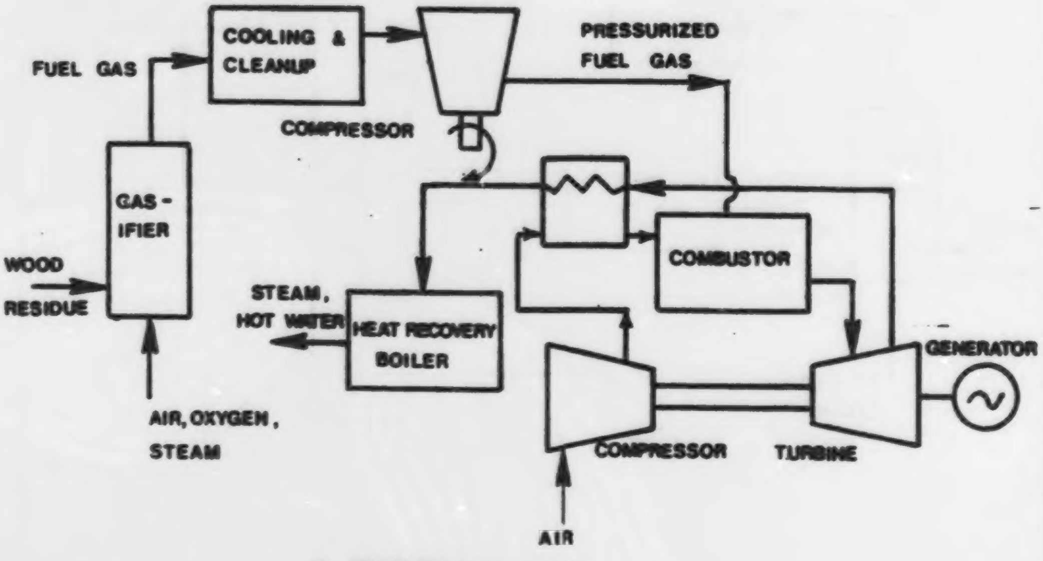
Several options exist for generating electricity with a fuel gas. The gas may be used after appropriate cleanup to fire a gas turbine or internal combustion engine directly. Systems incorporating an internal combustion engine are near commercial in status. No experience has been gained with gas turbines fired with wood gas, although tests have been conducted with fuel gas produced by a coal gasifier. Systems may be designed to generate steam in a gas-fired boiler and subsequently expand this steam in a conventional turbine or steam engine. These systems can be considered as near commercial to commercial technology, based on the status of gasifiers. Down stream of the fuel gas burner, the boiler and prime mover are well-proven commercial equipment. Operation of a fuel cell with gas produced by a gasifier has not been demonstrated, although a unit of several megawatts has been run with reformed naptha gas. Wood-derived gas would have to be reformed to produce a gas with a composition appropriate for fuel cell operation.

Figure III-9 presents cycle configurations for direct use of fuel gas in a turbine. Not included are indirectly fired turbines, since these are discussed with combustor conversion systems. Replacing the turbine prime mover with an internal combustion engine leads to the system shown in Figure III-10.

The cycles for gas turbines incorporate a compressor to raise the pressure of the fuel gas to the turbine inlet pressure. This is the only route possible with existing non-pressurized gasifiers. With development of commercial pressurized gasifiers, systems for generating power would not need an additional fuel-gas compessor. Such gasifiers are presently being studied in laboratory tests with air, oxygen, and steam injection.

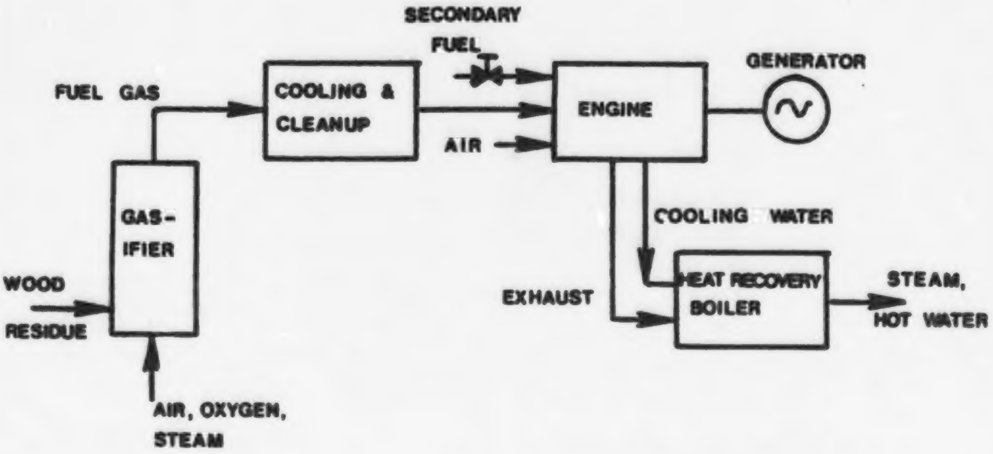


A. SIMPLE CYCLE

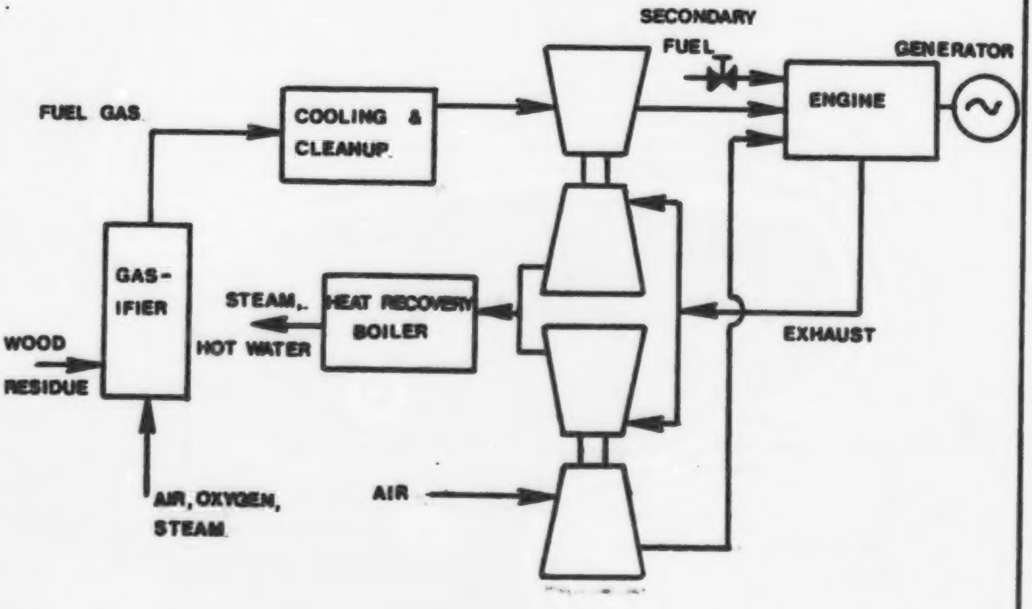


B. REGENERATIVE CYCLE

FIGURE III-9 GAS TURBINE TOPPING CYCLES WITH A GASIFIER



A. NATURALLY ASPIRATED SPARK IGNITION OR DUAL FUEL ENGINE



B. TURBO CHARGED SPARK IGNITION OR DUAL FUEL ENGINE

FIGURE III-10 INTERNAL COMBUSTION ENGINE TOPPING CYCLES WITH A GASIFIER

Cycles for internal combustion engines can be designed with or without fuel gas compression. For naturally aspirated engines, the flow of air into the gas is induced by the pressure differential developed by the engine suction. Turbocharged units have higher output capacities. However, problems are being encountered with blade deposition on the turbocharger. Engines could also be run with a pressurized gasifier.

Systems firing fuel gas to raise steam for expansion through a turbine or engine are of the same design as for conventional steam boilers. In this case, the steam boiler is replaced by two-stage conversion, firstly to fuel gas, secondly to heat. The efficiency of conversion of wood energy to steam for a gasifier/boiler combination will be about 60 to 68%, since the gasifier will convert 75 to 85% of incoming energy to hot dirty gas, and the boiler will convert about 80% of this energy to steam. This efficiency is comparable to a conventional hog fuel boiler run on wet wood, which will have an efficiency of about 65%.

The most sophisticated system for a gasifier uses the produced gas, after composition modification, to drive a fuel cell. Function of a fuel cell relies on delivery of hydrogen to the anode, and oxygen to the cathode in the presence of an electrolyte. Gas produced by a gasifier can be used in a fuel cell after cleaning and reforming to produce a gas high in hydrogen, and containing diluents such as CO, CO_2, and N_2, depending on the electrolyte used in the fuel cell. The higher the hydrogen concentration, the greater the capacity of the fuel cell. A simple schematic diagram for a gasifier/fuel cell system is given in Figure III-11.

b) General System Characteristics

Systems of near commercial or commercial status are those with either an internal combustion engine, or a gas-fired boiler and steam turbine/engine. Of these two systems, the gasifier/internal combustion engine combination is the more attractive since overall conversion efficiencies are higher, and the system is simpler and will require fewer controls and less maintenance. Another advantage in Canadian installations is that the system is not governed by the boiler code, enabling completely automated operation.

The exhaust from an internal combustion engine can be connected to a heat recovery boiler to supply medium pressure process steam. In most cases a gasifer/boiler combination with steam turbine/ engine would be less attractive than a

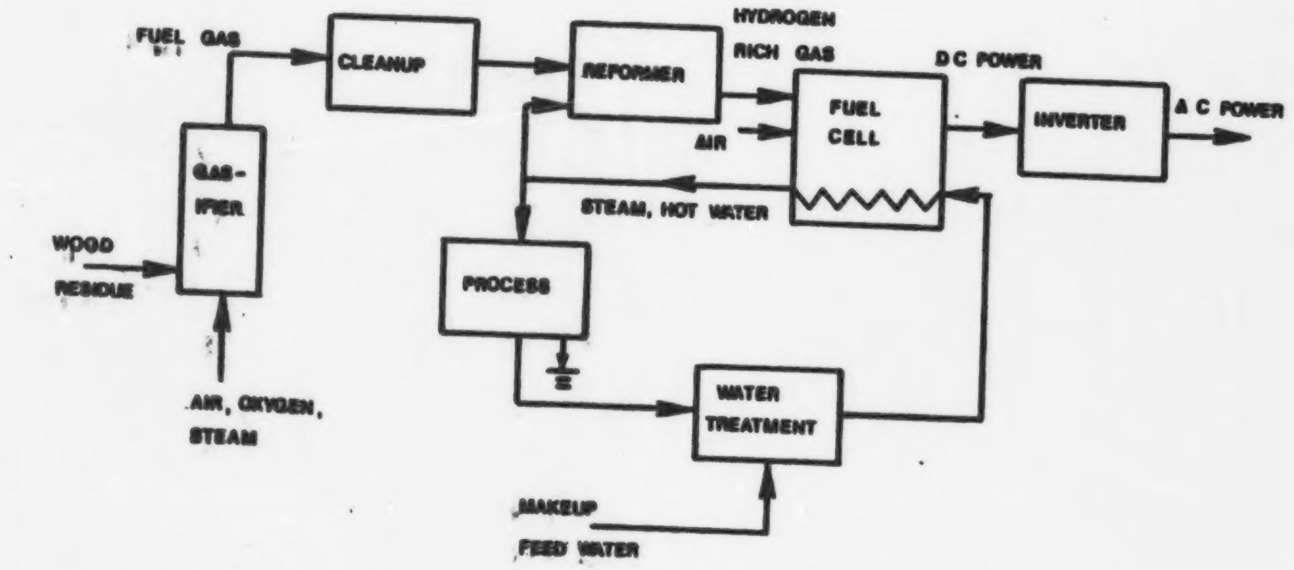


FIGURE III-11 FUEL CELL TOPPING CYCLE WITH A GASIFIER

conventional wood-fired steam boiler for steam raising. The most suitable circumstances for use of a gasifier to fire a boiler is where a gasfired boiler already exists.

A potential alternative to an internal combustion engine for a gasifier based system is a gas turbine. At present, no proven commercial turbine equipment is available for operation on low heating value gas derived from wood. Some tests and demonstrations of turbines have been done with low and medium heating-value clean gas.

Internal combustion engines are available in sizes mostly less than 2,000 kW. In this range, internal combustion engines are less expensive than comparable gas turbines. Internal combustion engines, naturally aspirated and turbocharged, have been operated on wood gas and generally function quite well. Some problems with tar deposits have been encountered when the gas is not cleaned sufficiently. Wet scrubbers have been used successfully for tar removal. Problems have also been encountered with regulation of the generator frequency because of variability of the gas composition and heating value. This problem is reduced by adequate mixing of the gas before it reaches the engine. A turbocharged engine design has the advantage of higher power output per unit engine displacement.

Internal combustion engines have efficiencies of conversion to electricity that, for their typical sizes, exceed values achieved by industrial gas turbines. Correspondingly, they have a higher value of the ratio of electricity to thermal energy. At full load, the E/T of diesel engines and regenerative gas turbines are about 1 to 1.1 and 0.6 to 0.7, respectively, including thermal energy in exhaust gas only and assuming a gas turbine pressure ratio of 6:1. Simple cycle gas turbines have an E/T of 0.35 to 0.40 for the same pressure ratio. This performance characteristic makes the internal combustion engine the most attractive prime mover for mills with little or no requirement for steam.

The last of the gasifier based conversion systems to consider is the one using a fuel cell. Since this technology is not presently commercial, it is not attractive for immediate use in forest products mills. This technology is most attractive for large central power systems or special military or space travel applications.

Fuel cells have short lifetimes (less than 5 years) and are very sophisticated power systems compared to alternatives considered in this study. They require not only a proven wood gasifier, but a proven gas reformer before a system could be developed. Both of these process units need further development before fuel cell systems could be attempted.

Fuel cells are very attractive because of their broad adaptability, the insensitivity of operating characteristics to scale and load and their modular design. Systems of several hundred kilowatts to thousands of megawatts are possible. Overall conversion efficiences of fuel cells exceed those of all other conventional internal combustion engines, gas turbine and steam turbine type power generating systems, with efficiencies of 40 to 45% for power plants exceeding 100 MW output.

4. Combustors (non-pressurized)

a) Cycle Configurations

Several systems are possible for generating electricity using a non-pressurized combustor. In two of these systems heat is exchanged from the hot combustion gas to a secondary working fluid. This secondary fluid may be either air to drive a gas turbine, or steam to drive an engine or turbine. Another system uses similar heat exchange to drive a Stirling engine which operates on an isolated working fluid. The final system uses advanced thermionics technology to generate electricity directly from the temperature difference between a hot combustion gas and a cooled reference collector.

Figure III-12 illustrates the systems for use of a combustor to indirectly fire a gas turbine in an open-cycle and a closed-cycle configuration. Both systems incorporate a regenerator for improved cycle efficiency, but ones without a regenerator can also be used. Heat recovery has been applied only to the primary working fluid (combustion gases), although additional heat recovery is also possible from the turbine exhaust after the regenerator. This gas stream will be at a lower temperature than the combustion gases exiting the heat exhanger.

Another indirectly-fired system uses hot combustion gases to raise steam which is subsequently expanded in either an engine or a turbine. This system accomplishes the same ends as a single steam boiler, except offers the potential for use of hot combustion gas for other purposes. The configuration of a system using a combination of a combustor and a heat recovery boiler is essentially identical to the one for a steam boiler previously discussed. These systems are proven technology, since both combustors and heat recovery boilers are well proven equipment.

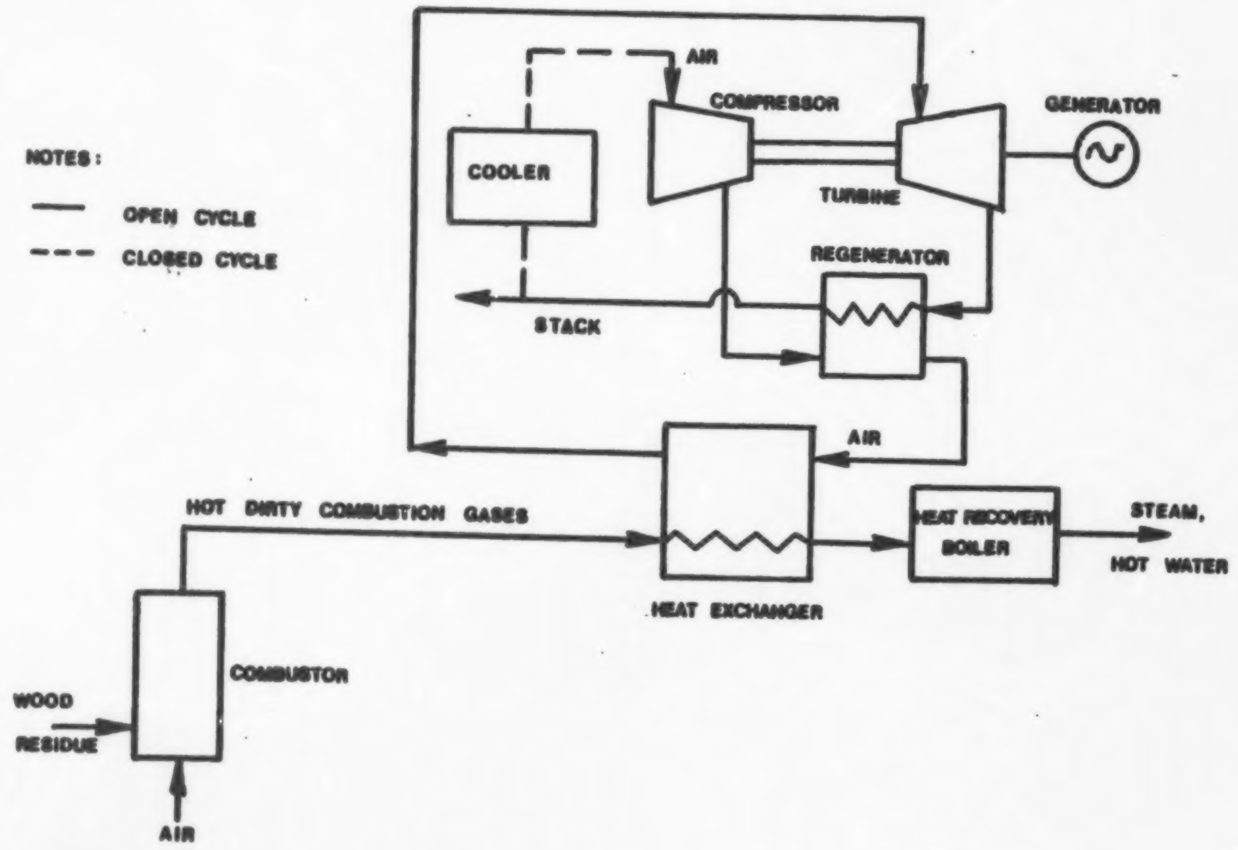


FIGURE III-12 INDIRECTLY FIRED GAS TURBINE TOPPING CYCLE WITH A COMBUSTOR

Stirling engines are external heat engines that use a separate working fluid selected specifically to optimize the operation of the engine. They can be coupled to either external combustors for operation with solid fuels, or integral combustors for operation with liquid or gaseous fuels. The configuration of a system using a Stirling engine coupled to an external combustor is shown in Figure III-13. Although the combustor shown is for operation on solid fuel, it could be replaced by a close-coupled gasifier and fuel-gas burner, using the fuel gas to generate heat needed by the engine. Steam can be generated with the residue heat in the gas exhausted from the Stirling engine by using a conventional heat recovery boiler.

At present, Stirling engines are only available commercially in sizes less than about 40 kW. The technology is available to build these engines in sizes up to in the area of 400 kW. Costs are presently high because of the lack of a sufficient market to enable mass production with associated economies of scale.

The most advanced of the identified combustor-based systems uses thermionic convertors to generate DC electricity directly from a heat soure. The DC power output may be converted to AC power either with an acyclic DC motor that drives a generator, or with a solid-state inverter. Thermionic devices are relatively simple devices that consist of a high-temperature electron emitter (1,000 to 1,300°C) and a low-temperature collector (380 to 600°C) separated by a small plasma-filled gap. Thermionic devices have been under development since the 1970's for application in large (greater than 100 MW) central power plants.

This development work has led to the construction of prototype designs suitable for topping applications with gas turbine combustors. Thermionic converter modules can range from 4 kW to 4 MW in size, and may be combined for very large outputs. The U.S. Department of Energy has a thermionics program aimed at near-term applications with gas turbine systems. The objectives of this program are to identify technically and economically attractive industrial applications, and to develop equipment.

A simple system layout for cogeneration of steam and electricity is shown in Figure III-14. Such systems are best suited to industries producing high temperature heat, and to utility power plants because of the very high emitter temperatures. The E/T of a utility-sized thermionic system with steam generation would be about 0.25 for a simple system with a ceramic heat exchanger for air heating by furnace gases, and about 0.18 with a metal rather than a ceramic heat exchanger.

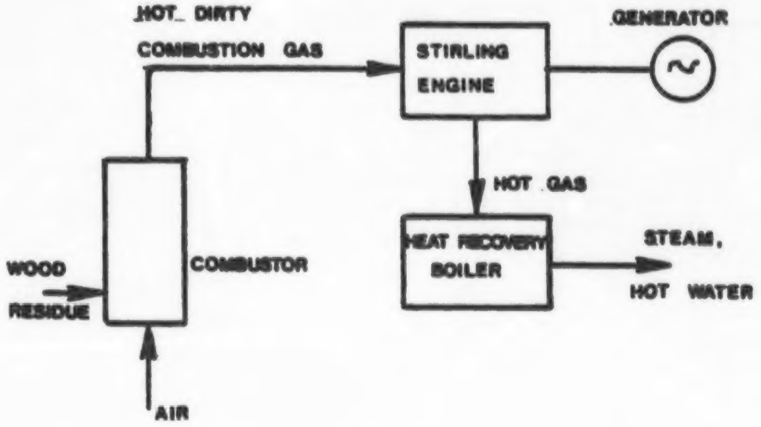


FIGURE III-13 STIRLING ENGINE TOPPING
CYCLE WITH A COMBUSTOR

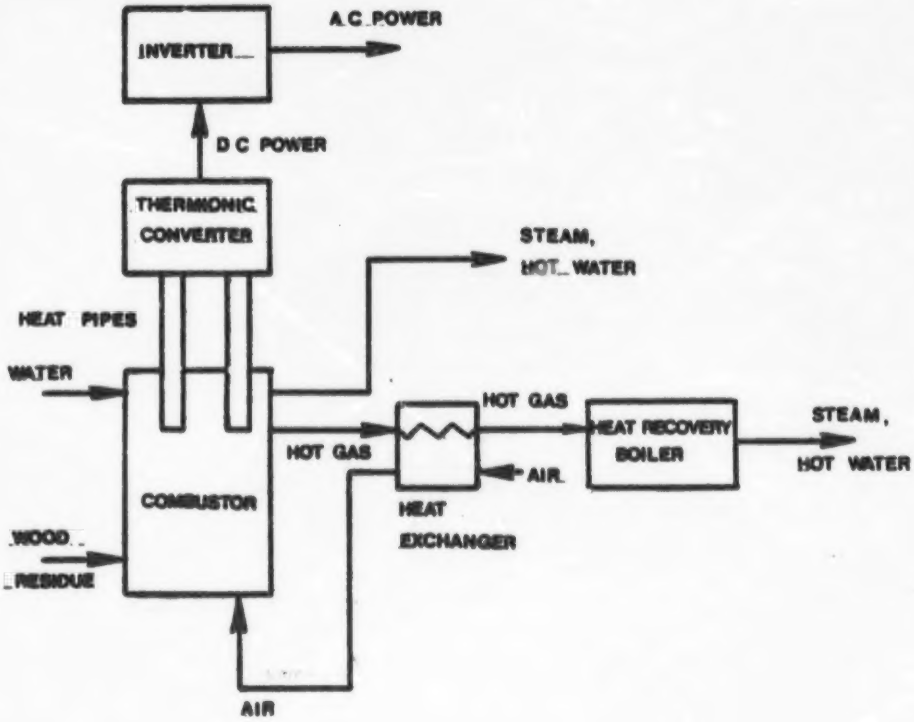


FIGURE III-14 THERMIONIC TOPPING CYCLE WITH A COMBUSTOR

Near-term industrial systems would have an E/T of less than 0.02 (Miskolczy et al).

b) General System Characteristics

Indirect firing of a gas turbine using air as the working fluid is technically feasible with presently available commercial equipment. Turbines operated in an indirect-firing configuration are best suited to combined cycle generation of electricity, with the heat removal for the gas turbine occurring upstream of the steam boiler. This configuration is preferable because only 5 to 6% of the energy in the hot combustion gases can be transferred to the air side of the heat exchanger. If the energy remaining in the combustion gas is used to generate process steam, then the system has an E/T of about 0.08, which is unsuitable for most forest product mills, including pulp and paper mills. With a combined cycle system including a condensing/extraction steam turbine the E/T increases dramatically to upwards of 2.0 when only the thermal energy in the extracted process steam is included, but reduces to 0.7 if the heat in the turbine exhaust is also included as available thermal energy. Although this E/T is very attractive for sawmills, a combined cycle plant is unattractive for most installations of less than 5 MW, and are more sophisticated than other alternatives for obtaining a high ratio of electrical to thermal energy.

Indirectly-fired turbines have the advantage of using a clean working fluid, thus minimizing or eliminating any problems with wear and corrosion of turbine parts. Because of the thermal inertia of the heat exchanger, precise speed control is more difficult than for a direct-fired unit, making the system best for supplying base load power. Start-up of the turbine must be done with either an electric motor (possibly the generator driven by backup power), or an auxiliary combustor.

The heat exchanger needed for the gas turbine system is near-commercial equipment. These heat exchangers may be constructed with ceramic or temperature-resistant metal materials, both of which result in high costs. Ceramics enable higher temperature operation, but limit the compressor and turbine pressure ratios to about 8:1 at temperatures up to 1,300°C.

A combination of a combustor and a packaged heat recovery boiler (for hot gas) for a steam power cycle is an alternative to a packaged boiler or a field erected boiler. Such a combination is best suited to installations where there is a requirement for hot gas, or when a modular system is desired to suit existing equipment. A separate combustor and boiler is an alternative for applications requiring less than 50,000 kg/h steam output. The performance characteristics of a combustor/boiler combination will be the same as for an integral steam boiler.

Modular construction of a combustor/boiler combination results in lower system costs than field erection and is very attractive for burning wet, coarse hog fuel. Several combustors are available that operate effectively on these fuels and can out-perform small grate-type integral steam boilers in their size range.

Stirling cycles have been under fairly continuous development for over ten years. The main advantages of Stirling engines are that they have the potential for higher conversion efficiences than gas turbine, steam turbine and internal combustion engines. Presently, conversion efficiencies for Stirling cycles are comparable to those for diesel engines and regenerative gas turbines.

Since a Stirling engine is externally heated, it can be connected to a combustor fired by wood residues. The working fluid is isolated from the heat supply, enabling clean operation of the engine and a very long service life for all moving parts.

Stirling engines are available commercially in limited supply in sizes less than about 40 kW. Although many studies have considered large Stirling engines suitable for industrial and utlity power plants, no units of suitable size are presently available. Development of these larger engines awaits commercialization of the small units (40 to 150 kW) for use as portable power plants and mechanical drives.

Stirling engines are attractive emerging technology for generation of electricity from wood because the engine is suited to an external combustor. Ratios of electrical to thermal energy are near 0.6 for current equipment, and are expected by development companies to increase to over 1.0 with improved units. The engines have better part-load behavior than internal combustion engines and gas turbines.

The key component controlling the technical feasibility of connecting a Stirling engine to a wood combustor is the heat exchanger that transfers heat to the working fluid, usually helium, from the combustion gases. These gases must be cleaned sufficiently to prevent fouling of the heat exchanger, which, on present units, has been designed for clean-burning fuels only. This heater runs at a temperature of about 700°C, a temperature that is compatable with gases from wood combustors, which are 1,100°C or less.

It is also possible to fire a Stirling engine with wood-derived fuel gas by utilizing a suitably designed combustor. This has the advantages of reducing the ash content of the gas as compared to a wood combustor, but reduces the overall conversion efficiency and may encounter problems with tar contained in the fuel gas.

Generation of electricity thermionically is at the research level and involves use of advanced heat-pipe technology, as well as the thermionic convertors themselves. This technology is not applicable to the forest products industry in the foreseeable future, but has potential for central power plants.

5. Pressurized Combustors

Although any wood combustor could, with appropriate design, be operated at elevated pressure, only fluid-bed combustors are normally considered for this type of operation. Fluid-bed combustors are attractive for pressurized operations because of their potential for a high grate heat release rate and the inherently good contact between fuel and oxidant in the turbulent bed.

Only fluid-bed combustors are considered in this section.

a) System Configuration

Figure III-15 shows a system configuration for a pressurized fluid-bed combustor driving a gas turbine. This type of system has been demonstrated by Combustion Power Inc. with a pilot plant operation. No commercial pressurized combustors are available presently. A conventional gas turbine with an auxiliary combustor for startup could be adapted to this system, as was done for the pilot test.

b) General System Characteristics

The Combustion Power pilot plant was designed primarily for municipal solid waste, but limited length runs were also conducted with hog fuel. The pilot plant had an electrical output capacity of 1 MW and incorporated a combustor that operated at 465 kPa and three stages of gas cleanup, the last stage being a granular filter. The turbine operated with an overall pressure ratio of 3.5:1.

Problems were encountered with prohibitive deposition of ash on the turbine blades. Since the test program the granular filter has been improved with the addition of an electrically charged grid and is available commercially. This unit

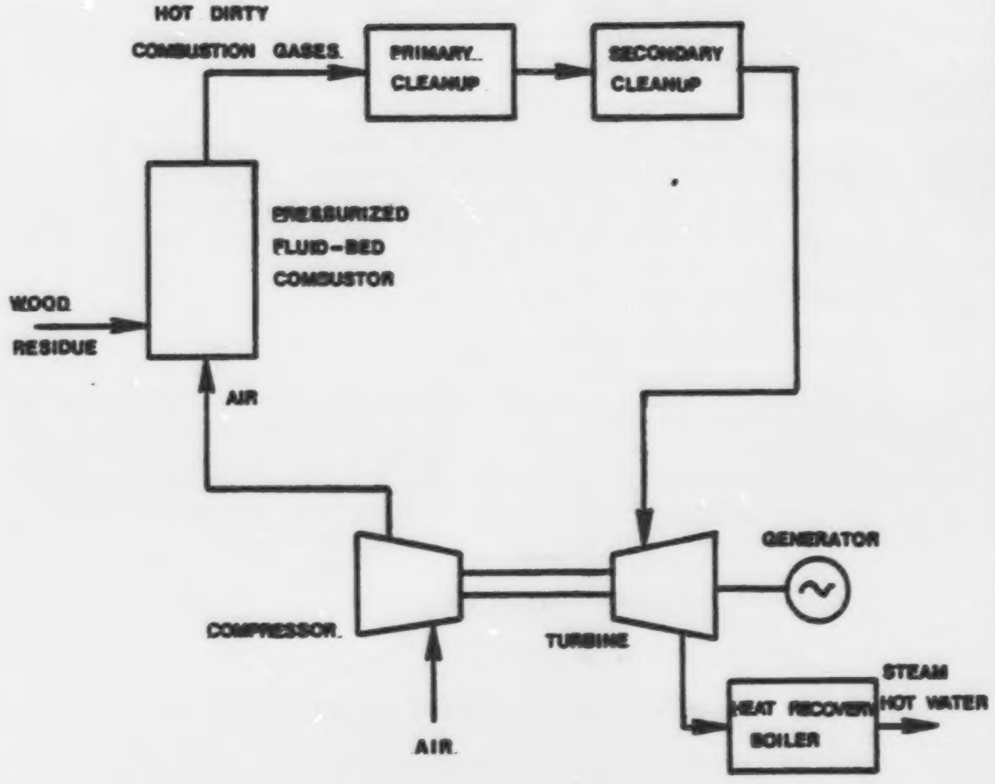


FIGURE III-15 DIRECT FIRED GAS TURBINE TOPPING CYCLE WITH A FLUID-BED COMBUSTOR

is capable of removal of particulate down to half micron particle size with a pressure drop of a few inches of water gauge. The improved performance of the refined granular filter would reduce the problem with deposits that was experienced with the pilot tests. Some fouling problems would undoubtedly be experienced with the fine unfiltered ash because of both deposition and corrosion by alkali metals that are present in the ash.

The conversion efficiency (wood to electricity) of the direct-fired turbine with a wood combustor used in the pilot plant was 11%. The upper limit on efficiency is about 16% for the turbine, based on 90% conversion in the combustor and a 18% simple cycle efficiency (Figure VI-7). The E/T of this system theoretically is near 0.22, and suitable for a pulp and paper mill requiring process steam. Higher E/T would be possible with a regenerative turbine.

The conversion efficiency and E/T could be improved by raising the pressure ratio of the turbine and, consequently, the combustor pressure. Demonstration of this technology is needed before any conclusions can be drawn concerning its technical feasibility. Obstacles to development are the problem of dependable feed of wood residues into high pressure reactors and ash deposition and corrosion of the turbine.

This type of direct-fired turbine would be best for base load because of the thermal inertia of the combustor, gas filter and gas ducting. Since the fluid-bed combustor has to operate at near constant temperature, and the turbine has to operate at constant speed, turbine output must be controlled using by-passes on the combustor inlet and outlet.

6. Pulverizers

No turbines exist for firing pulverized waste directly in an integral combustor in a manner comparable to firing fossil fuels. Some research with turbines fired on pulverized coal has been conducted. In these tests serious problems were encountered with deposition of ash and corrosion of turbine parts. Although the ash content of coal is 5 to 15 times greater than the ash content of wood, similar fouling problems will occur. This is demonstrated by results from tests with a turbine connected to a pressurized fluid-bed combustor.

7. Liquefaction Units

Very low conversion efficiencies, probably on the order of 8% or less, would result from production of a liquid fuel from wood and subsequent generation of electricity with a gas turbine or internal combustion engine. In addition, these systems would be logical only if the liquid fuel was the primary product. Predicted costs for liquid fuels from wood exceed those of fossil fuels, so would presently by very unattractive for power generation.

8. Bottoming Heat Recovery Boilers

a) System Configuration

Figure III-16 presents a configuration for a bottoming Rankine cycle where an organic fluid is used as the working fluid. Fluids applicable to these cycles are: halocarbons (refrigerants), hydrocarbons, ammonia, and several other organic and water-organic fluids.

The working fluid behaves much like steam in a traditonal Rankine cycle, vapourizing in the boiler, expanding in a single-stage (common) or multiple-stage turbine and condensing prior to return to the boiler.

b) General System Characteristics

In an extensive survey of operating organic Rankine cycle systems, Curran identified 2,150 operational engines from 20 different manufacturers using 16 different working fluids. These systems had power outputs that varied from 0.1 kW to 1,120 kW in applications including driving water pumps, compressors and electric generators. The largest manufacturer of these systems, Ormat Turbines of Yavne, Israel, has units installed world wide in sizes of 1 to 6 kW with a combined operating time of some 18 million hours.

Most of the operating systems are for research and development purposes, and have 100 to 1,000 hours of operating time.

Organic fluid Rankine cycles can have higher efficiencies than steam cycles for heat source temperatures below 400°C (Curran). The main target for these systems is heat recovery from waste heat streams at temperatures of 100 to 400°C. The efficiency of these organic Rankine cycles in this temperature range falls between about 10% and 30%. Operating pressures range from 200 kPa to 3,400 kPa.

A serious disadvantage of organic fluids is their toxicity. Leakage during handling or from an operating system may be hazardous to the health of exposed workers and pose a safety problem. There is also a problem with disposal of the fluids.

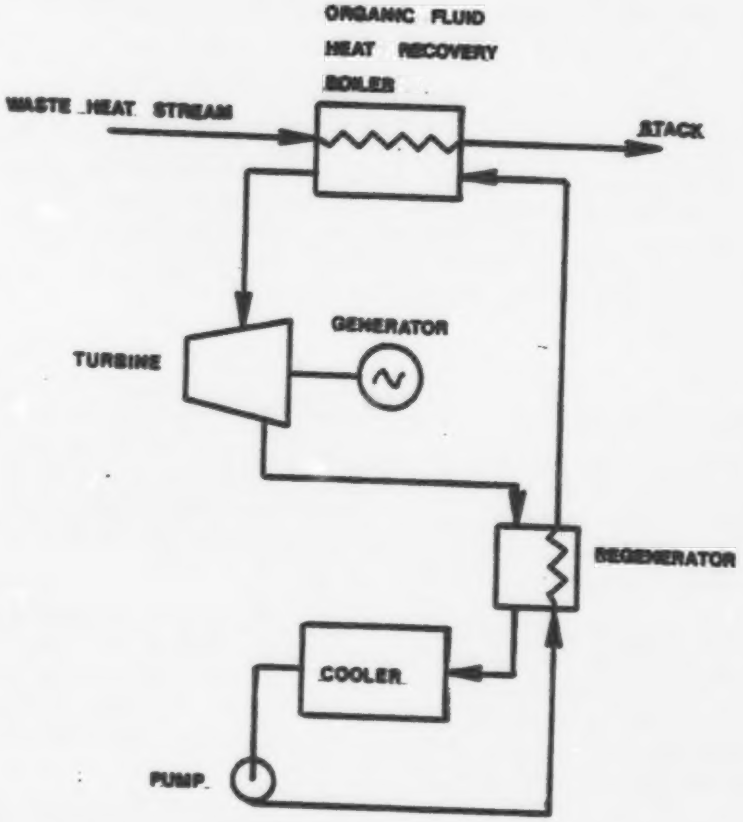


FIGURE III-16 VAPOUR TURBINE BOTTOMING RANKINE CYCLE USING AN ORGANIC FLUID

The published literature reports few operating problems related to the behaviour of the fluid and exposed metal parts, although this may be due to lack of continuously-operated systems.

A study by Resource Planning Associates found that organic Rankine cycles for generation of electricity from waste heat suited chemical and petroleum refining industries better than the pulp and paper industry because of the abundance of recoverable waste heat in these industries. This technology looks unattractive for all forest products industries but the pulp and paper industry, where there may be circumstances warranting improved waste heat recovery from hot water streams.

C. Rationale for Selecting an Electricity Generating System

A total of nineteen systems for generating electricity from wood residues were considered in this study. This is certainly not all the systems, since some of the emerging technologies were neglected, but includes all systems that, technically, merit consideration as options for the forest product industry.

Because of the profusion of options, some rationale is needed for selecting the best possible choice for a particular situation. The following discussion concentrates on, firstly, the performance characteristics of the systems to eliminate those unsuitable for the forest products mills, and secondly, operational considerations that make some promising candidates better than others.

The rationale presented here are generalized and are intended to allow screening of the available systems for identification of a limited number of candidates. These should then be subjected to an economic analysis which accounts for site specific factors. The costs of systems components are given in the body of the report, where available from suppliers.

1. System Performance Characteristics

A summary of the performance characteristics of all the systems considered in this study is given in Table III-1 along with a brief treatment of several merits of the systems that can be used as guidelines for selection. This table states the forest industry suited to each system based solely on the match of the electrical-to-thermal energy ratio to the mill requirement. Discretion must be excercised in interpreting this suitability assessment, since numerous site specific factors may reduce the importance of the E/T ratio.

Table III-1

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
STEAM BOILER:					
-**STEAM TURBINE**		-proven equipment with many options -good operating flexibility -worker familiarity -moderate cost -packaged units available	-requires trained operators; -need water source -need use for steam -high maintenance	-none	-Pulp and paper mills -Sawmills with dry kilns -Plywood and veneer mills
SYSTEM STATUS	-commercial				
POWER OUTPUT (kW)	->300; >1,500 typical				
E/T	-0.04-0.45				
BOILER CODE	-applicable				
-**STEAM ENGINE**		-lower steam rates than turbines below 2 MW -good for variable speed and slow speed applications -good torque characteristics -low maintenance -worker familiarity -good operating flexibility like turbines	-limited availability, sizes and options -requires trained operators; non-automated -need water source -need use for steam	-steam engine	-Pulp and paper mills -Sawmills with dry kilns -Plywood and veneer mills
SYSTEM STATUS	-commercial				
POWER OUTPUT (kW)	-25 - 3,000; up to 20,000 possible				
E/T	-0.04 - 0.45				
BOILER CODE	-applicable				
GASIFIER AND PYROLYZER:					
-**FUEL CELL**		-no moving parts in fuel cell -low emissions -operating flexibility -modular construction -high efficiency over-wide load range	-unproven -no commercial systems available -very sophisticated unfamiliar technology -high cost	-gasifier -gas reformer -fuel cell	-Unknown
SYSTEM STATUS	-conceptual (pilot operating on naptha)				
POWER OUTPUT (kW)	->20				
E/T	-1 - 1.2				
BOILER CODE	-not applicable				

Table III-1 (continued)

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
GASIFIER AND PYROLYZER:					
-GAS TURBINE		-suitable for automation -compact -low labour requirement	-best for base load applications -poor part-load performance -operational inflexibility -fouling of turbine -unproven, limited availability -high maintenance	-gas cleanup -turbine combustor -control system	-possibly sawmills -specialty products mills
SYSTEM STATUS	-conceptual/ research				
POWER OUTPUT (kW)	-500 - >100,000				
E/T	-0.2 - 0.8				
BOILER CODE	-not applicable				
-INTERNAL COMBUSTION ENGINE		-suitable for automation -low labour requirement -moderate maintenance -fair equipment availability	-possible problem with engine control due to fluctuations of gas quality -tar and particulate deposits in engine -scrubber water disposal -lube oil contamination	-gas cleanup -control system	-sawmills -specialty products mills
SYSTEM STATUS	-demonstration/ commercial				
POWER OUTPUT (kW)	-100 - >2,000				
E/T	-0.5 - 0.9				
BOILER CODE	-not applicable				
-STEAM TURBINE		-tars incinerated, maximizing energy recovery from fuel gas -gas available for other uses -others as per steam boiler	-limited equipment availability -more complex than combustor/boiler combination -others as per steam boiler	-gas burner -steam engine	-pulp and paper mills -sawmills with dry kilns -plywood and veneer mills
SYSTEM STATUS	-demonstration/ commercial				
POWER OUTPUT (kW)	-300 - 8,000				
E/T	-0.4 - 0.45				
BOILER CODE	-applicable				
-STEAM ENGINE					
SYSTEM STATUS	-demonstration commercial				
POWER OUTPUT (kW)	-25 - 3,000				
E/T	-0.04 - 0.45				
BOILER CODE	-applicable				

Table III-1 (continued)

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
COMBUSTOR:					
-GAS TURBINE		-clean gas for turbine -suitable for automation -if closed cycle, good part load performance and efficiency -low to moderate turbine maintenance	-need use for hot gas or steam -thermal inertia poses control problem -best for base load, steady operation -limited availability of heat exchanger for high temeperature	-heat exchanger -control system	-pulp and paper mills
SYSTEM STATUS	-pilot				
POWER OUTPUT (kW)	->500				
E/T	-approx. 0.08				
BOILER CODE	-not applicable; with combined cycle is applicable				
-STIRLING ENGINE		-engine uses isolated working fluid -long engine life -suitable for automation	-ash fouling of heater in stirling engine -thermal inertia poses control problem -worker infamiliarity -limited availability of engine -small output	-gas cleanup -Stirling engine	-sawmills -specialty products mills
SYSTEM STATUS	-prototype/ demonstration				
POWER OUTPUT (kW)	-<40, presently				
E/T	-0.6 - 1.0				
BOILER CODE	-not applicable				
-STEAM TURBINE		-hot gases available for other uses -flexible modular system structure -others as per steam boiler	-thermal inertia poses control problem -others as per steam boiler	-steam engine	-pulp and paper mills -sawmills -plywood and veneer mills
SYSTEM STATUS	-commercial				
POWER OUTPUT (kW)	-300 - 11,000				
E/T	-0.04 0.45				
BOILER CODE	-applicable				
-STEAM ENGINE					
SYSTEM STATUS	-demonstration/ commercial				
POWER OUTPUT (kW)	-25 - 3,000				
E/T	-0.04 - 0.45				
BOILER CODE	-applicable				

Table III-1 (continued)

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
COMBUSTOR:					
-THERMIONICS		-no moving parts in converter -good for combined cycle	-unproven -no commercial systems available -high cost -unfamiliar technology -need use for hot gases	-all system components	-unknown, E/T poor at present for all mills
SYSTEM STATUS	-research				
POWER OUTPUT (kW)	-4 - >100,000				
E/T	-0.02, presently, 0.2 - 0.25, future				
BOILER CODE	-not applicable				
PRESSURIZED COMBUSTOR:					
-GAS TURBINE		-suitable for automation -compact	-thermal inertia of combustor poses control problem -best for base load -ash fouling and wear of turbine -no commercial systems available	-gas cleanup -gas turbine -fuel feed system	-sawmills -specialty products mills
SYSTEM STATUS	-research				
POWER OUTPUT (kW)	-1,000 tested				
E/T	-0.2 - 0.8				
BOILER CODE	-not applicable				
PULVERIZER:					
-SOLID FUEL TURBINE		-simple system -suitable for automation	-high power required for pulverizing -ash fouling and wear of turbine -no commercial systems available	-all system components	-sawmills -specialty products mills
SYSTEM STATUS	-conceptual				
POWER OUTPUT (kW)	-unknown				
E/T	-0.2 - 0.8				
BOILER CODE	-not applicable				

Table III-1 (continued)

PERFORMANCE CHARACTERISTICS AND SELECTION GUIDELINES FOR GENERATING SYSTEMS

PRIMARY CONVERSION/ PRIME MOVER	CHARACTERISTICS	ADVANTAGES	DISADVANTAGES	EQUIPMENT NEEDING IMPROVEMENT	SUITABLE FOREST INDUSTRY
LIQUEFACTION:					
-GAS TURBINE, INTERNAL COMBUSTION ENGINE		-conventional prime movers	-unproven -very low conversion efficiency -high cost	-all system components	-unknown
SYSTEM STATUS	-conceptual				
POWER OUTPUT (kW)	-N/A				
E/T	-N/A				
BOILER CODE	-not applicable				
HEAT RECOVERY BOILER:					
-ORGANIC VAPOR TURBINE		-recovery of waste heat -higher efficiency than steam below 400°C -best for recovery at 100°C - 300°C	-toxic fluids -problem to handle and dispose of fluids -limited availability of systems -degradation of fluids -worker unfamiliarity	-all system components	-pulp and paper mills
SYSTEM STATUS	-demonstration				
POWER OUTPUT (kW)	-0.1 - 1,000				
E/T	-does not apply				
BOILER CODE	-probably does apply				
-STEAM TURBINE		-SEE STEAM BOILER			
SYSTEM STATUS	-commercial				
-STEAM ENGINE					
SYSTEM STATUS	-commercial				

Several of the systems included in Table III-1 use emerging technology that is unlikely to suit the needs of forest products mills in the foreseeable future. Removing these advanced or inappropriate systems, a condensed list of electricity generating systems was developed. Using the values of E/T for sawmills, pulp and paper mills, and specialty products mills, the candidate systems were matched to these mill groups (Table III-2). Bottoming organic Rankine cycles are excluded from this analysis because of the inappropriateness of the E/T ratio for this type of system. Organic Rankine cycles are best for industries having large amounts of waste heat at low temperatures. This characteristic makes them most appropriate for pulp and paper mills.

Many of the systems are best suited to a certain range of power output because of factors related to manufacture, operating principle, efficiency, cost, etc. Typical power output capacities of the candidate systems for the forest products industry are presented in Figure III-17. These ranges are based on information from suppliers concerning sizes of prime movers and on capacities of commercial wood conversion systems. This data shows the attractiveness of steam engine systems in sizes up to several hundred kilowatts and possibly higher. In this size range, a gasifier/I.C. engine combination is the prime candidate. With further commercial development, Stirling engines may also be an attractive alternative.

The most promising developing prime mover for mills with a low demand for thermal energy is the Stirling engine. This engine presently is more suitable than gas turbines coupled with either a gasifier or a pressurized combustor because the moving parts of the engine are not in contact with hot, dirty gases.

2. Operational Considerations

Traditionally, electricity is generated at mills using large steam boilers and either back-pressure or condensing/extraction turbines. Consequently, steam-based generaton will usually be the system to which all alternatives are compared.

One of the main operational considerations for steam-based electricity generation is the manpower requirement and cost. The manning of these systems is dictated by provincial boiler codes summarized in abridged form in Table III-3. All boilers suitable for power generation are governed by the provincial boiler acts. These power systems must be operated and supervised by accredited personnel.

The remaining operating considerations depend on the type of electricity generating system selected. Most of these considerations are covered in the discussion of each of the unit processes.

Table III-2

COMPATIBILITY OF CANDIDATE POWER SYSTEMS WITH SAWMILLS, PULP AND PAPER MILLS, AND SPECIALTY PRODUCT MILLS BASED ON THE ELECTRICAL TO THERMAL ENERGY RATIO

	MILL TYPES		
	SAWMILLS	PULP AND PAPER, PLYWOOD	SPECIALTY PRODUCTS
E/T of Mills	0.1 - 1.0	0.1 - 0.2	0.3 - 1.0
Candidate Systems:			
boiler/steam turbine	Fair	Good	Poor
/steam engine	Fair	Good	Poor
gasifier/gas turbine	Good	Poor	Good
/I.C. engine	Good	Poor	Good
/steam turbine	Fair	Good	Poor
/steam engine	Fair	Good	Poor
combustor/gas turbine	Poor	Fair	Poor
/Stirling engine	Good	Poor	Good
/steam turbine	Fair	Good	Poor
/steam engine	Fair	Good	Poor
pressurized combustor/ gas turbine	Good	Poor	Good

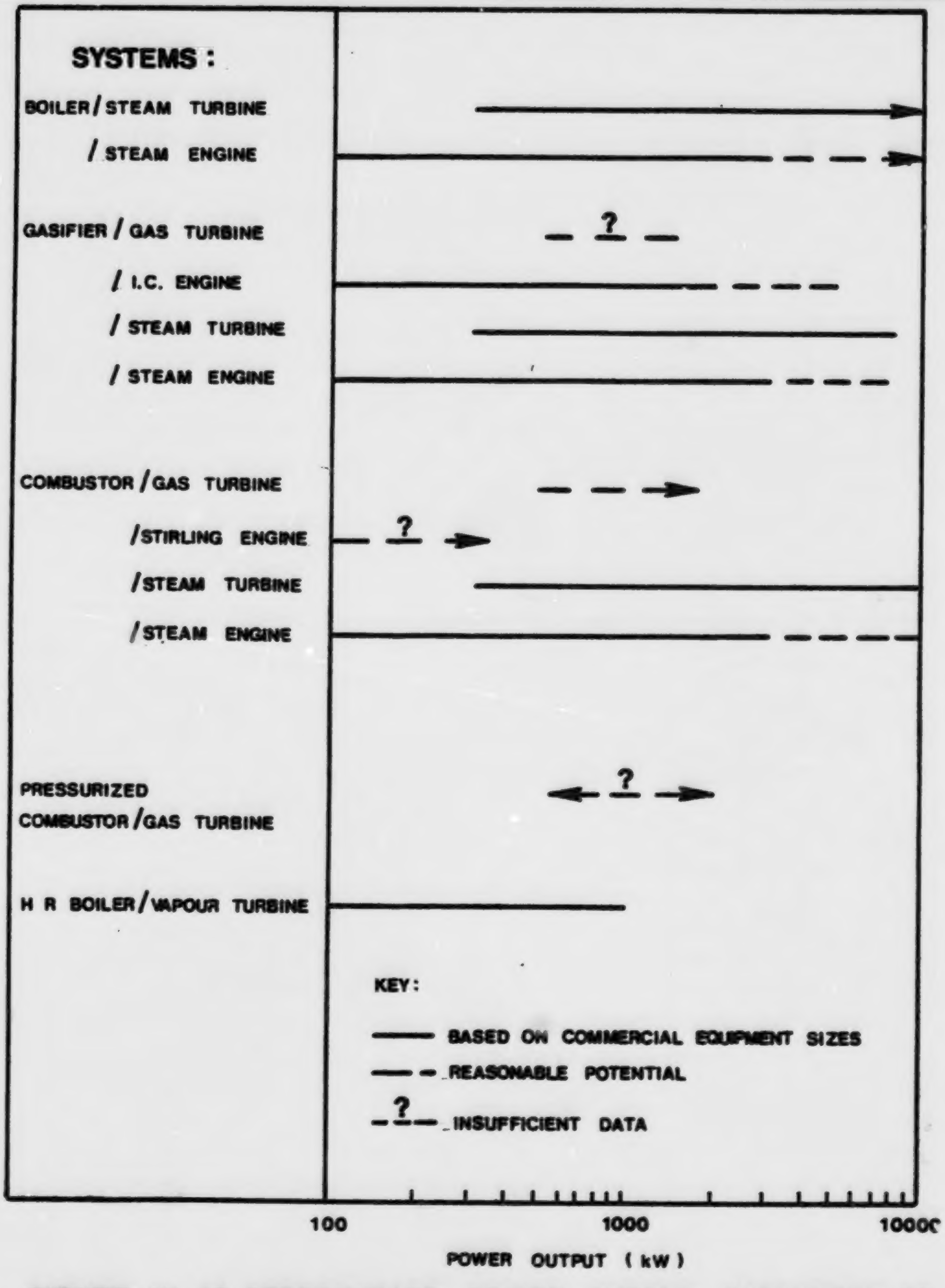


FIGURE III-17 APPROXIMATE POWER OUTPUT CAPACITIES OF SYSTEMS FOR WOOD RESIDUE CONVERSION

Table III-3

ABRIDGED SUMMARY OF PROVINCIAL BOILER AND PRESSURE VESSEL ACTS

PROVINCE	APPLICATIONS COVERED BY ACT	EXEMPTIONS FROM ACT
British Columbia:	-All boilers, except as exempted. -Boiler is a vessel in which a gas vapour is generated or a liquid is put under pressure by heating.	-Steam boiler of 2 hp or less than 3 ft^3 volume. -Vented hot water heating system. -Steam boiler of 15 psi or less, and of 3 hp or less capacity. -Hot water boiler of 160 psi or less and 250°F or less.
Alberta:	-All boilers, except as exempted. -Boiler is a vessel in which a vapour is generated or a liquid is put under pressure by heating.	-Boiler of 10 kW or less capacity, forming part of a power plant. -Boiler of 20 kW or less capacity, installed in heating plant. -Boiler for vented hot water heating system. -Boiler in which vapours are at 103 kPa or less or liquids are at 1,100 kPa and 121°C or less.
Saskatchewan:	-All boilers, except as exempted. -Boiler is a vessel in which steam or hot water is produced under pressure.	
Manitoba:	-All boilers, except as exempted. -Boiler is a vessel in which steam is generated or a liquid is produced under pressure.	-Boilers that develop less than 3 hp using steam or hot water.

Table III-3 (continued)

ABRIDGED SUMMARY OF PROVINCIAL BOILER AND PRESSURE VESSEL ACTS

PROVINCE	APPLICATIONS COVERED BY ACT	EXEMPTIONS FROM ACT
Ontario:	-All boilers, except as exempted. -Boiler is a vessel to generate steam, or to heat water under pressure.	-Boiler with steam at 15 psi or less, or water at 250°F or less, and rated at 1,000,000 BTU/h or less and not connected to another boiler or connected and having a rating of 5,000,000 BTU/h. -Boiler for vented hot water heating system. -Boiler with steam exceeding 15 psi or water exceeding 250°F with a rating of 500,000 BTU/h or less when alone or rating of 1,700,000 BTU/h or less when connected to others. -A coiled tube boiler containing steam at a pressure of more than 15 psi and having a water content of 75 imperial gallons or less.
Quebec:	-All pressure vessels, except as as exempted.	-Steam, hot water, thermal liquid boiler of 15 psi or less with heating surface not exceeding 30 ft^2. -Vessels with a gas under pressure not exceeding 15 psi. -Vessels with a gas vapour or liquid having a capacity not exceeding 1.5 ft^3.
New Brunswick:	-All heating plants and power plants, except as exempted. -Boiler is a vessel in which steam is generated, or hot water is produced under pressure having a capacity of more than 3 ft^3 or area exceeding 20 ft^2.	-Boilers with steam at 15 psi or less, or water at 160 psi and 250°F or less, and capacities of 5,000,000 BTU/h or less. -Boilers with steam exceeding 15 psi, or water exceeding 160 psi and 20°F and capacities of 2,000,000 BTU/h or less.

Table III-3 (continued)

ABRIDGED SUMMARY OF PROVINCIAL BOILER AND PRESSURE VESSEL ACTS

PROVINCE	APPLICATIONS COVERED BY ACT	EXEMPTIONS FROM ACT
Nova Scotia:	-All steam boilers, except as exempted. -Boiler is a vessel in which steam is generated for power or heating purposes.	-Boilers with steam at 15 psi or less.
Prince Edward Island:	-All boilers, except as exempted. -Boiler is a vessel in which steam or vapour is generated, or hot water or liquid is produced or put under pressure by heating. -Any plant burning solid fuel shall be under continuous supervision.	-Boilers of 1,500,000 BTU/h or less and steam pressure of 15 psi or less, hot water of 160 psi, or 250°F or less.
Newfoundland:	-All boilers, except as expempted. -Boiler is any vessel in which gas or vapour may be generated, or liquid may be put under pressure by the application of heat.	-A pressure vessel operated at 15 psi or less. -Boilers, other than a miniature boiler, with a heating surface of 30 square feet or less (approximately 100,500 BTU/h).

IV WOOD CONVERSION UNITS

A. General

The discussion of system configurations given in Chapter III identified the fundamental processing units for generating electricity from wood residues: Primary conversion unit; Secondary conversion unit; and Prime mover unit. This chapter presents a discussion of the options available for wood conversion units.

All wood conversion units that convert raw wood to a useable form of energy, such as steam, hot gas, fuel gas or liquid fuel, were considered. These were classified under the following major headings: Steam Boilers; Combustors; Gasifiers; Pyrolyzers; and liquefaction Units. Under each of these headings are considered the various distinct unit designs that are presently available or under development.

The following sections summarize the performance characteristics of the various wood conversion units, drawing information from published literature and replies from manufacturers and users of commercial equipment. The summarized data and accompanying discussion are sufficient to allow the reader to assemble one or more possible system designs for generating electricity and compare their inherent advantages and disadvantages, and to estimate, very approximately, the system costs.

A thorough discussion of the various wood conversion units, by manufacturer, is available in the most recent edition of "An Evaluation of Wood Waste Energy Conversion Systems 1980", ENFOR C-111, by B.H. Levelton & Associates Ltd. A general treatment of the use of wood fuels is presented in the report, "Decisionmaker's Guide to Wood Fuel for Small Industrial Energy Users" by the Solar Energy Research Institute. A description of gasifier systems can be obtained from the report, "A Survey of Biomass Gasification Vol. I, II and III", also by the Solar Energy Research Institute. The reader is referred to the above documents or directly to the manufacturers for specific details about commercial units and locations of installations, information not contained in this report.

B. Types and Definitions of Wood Conversion Units

The following general definitions were used to classify wood conversion units into groups having a similar type of operation and form of discharged working fluid. The working fluid in this discussion is the fluid discharged from the conversion unit and used either directly, or indirectly (after heat is exchanged with a secondary working fluid) to drive a prime mover.

Conversion Units:

Steam Boiler - Conventional solid fuel boiler for generation of steam. Includes packaged, modular and field erected boilers.

Combustor - Device for complete combustion of fuel and generation of hot combustion gas/air mixture. Includes suspension burners, single chamber combustors, multiple chamber combustors, and fluid-bed combustors.

Gasifier - Device for partial combustion of fuel and generation of fuel gas of low (3.7 - 6.5 MJ/m^3) or medium (7.4 - 11 MJ/m^3) heating value. Includes fixed-bed (updraft and downdraft), stirred bed and fluid-bed gasifiers.

Liquefaction Units - Device for chemical decomposition of fuel, directly or indirectly for production of a liquid fuel and chemical co-products. Fuels may include synthetic wood-derived liquids from direct liquefaction, or alcohols from indirect liquefaction.

Pyrolyzer - Device for chemical decomposition of fuel by heating for production of char, liquid tars and oils, and fuel gas.

C. Combustors

Combustors may be of suspension, fluid-bed, single chamber or multiple chamber design. Each design has fairly unique operating characteristics and conversion performance.

To assist in the preliminary selection of a combustor appropriate to a given situation, a brief summary of data that are discussed later in this section are presented in Table IV-1. These data show some of the attributes of the available commercial combustors. For example, single chamber combustors are of limited capacity, typically less than 6 GJ/h output, and, therefore, at 15% conversion efficiency to electricity could generate up to only 170 kW, assuming equipment was available for utilization of the thermal energy in a prime mover. Suspension burners are commonly available in sizes up to 85 GJ/h, although designs for larger units are available. This range of sizes is less than that available for both fluid-bed and multiple chamber combustors.

Table IV-1 gives typical ranges of costs for units including a feed system and basic controls. The ranges of the costs are a

Table IV-1

TYPICAL OPERATING PERFORMANCE AND COSTS FOR COMBUSTORS

Criteria	Suspension Burner	Single Chamber Combustor	Multiple Chamber Combustor	Fluid-Bed Combustor
Fuel: Size (cm)	<0.1[a] <0.3 X 0.3 X 2[b]	<5	<8	<8
Moisture Content (%, wb)	<10[a] <15[b]	<50	<60	<65
Output Capacity (GJ/h)	5 - 85	<6	1 - 150	2 - 130
Turndown	5 : 1	N/A	4 : 1	2.5 : 1
Thermal Efficiency [c] (%)	>95	95	95	95
Excess Air (%)	15 - 35	N/A	10 - 45	50 - 100
Hot Gas Temperature [d] (°C)	600 - 1,200	600 - 1,100	900 - 1,200	600 - 1,100
Particulate Loading (g/m^3; 15°C, 12% CO^2)	2.5[e]	<0.25	0.23 - 0.46	N/A
Equipment Cost ($/GJ/h)	10 GJ/h - 7,800[a] 13,600[b] 60 GJ/h - 2,000[a] 3,800[b]	0.5 GJ/h-53,000 5 GJ/h-16,000	10 GJ/h-20,000 140 GJ/h-10,000	10 GJ/h-37,000 100 GJ/h-8,500

[a] Suspension burner only
[b] Suspension burner with furnace chamber
[c] Allows for unrecoverable losses only - combustibles in ash; ash-sensible heat; wall losses
[d] Upper temperature limited by ash-softening behaviour
[e] At burner assuming all ash emitted - ash content = 2%, db

consequence of economics of scale as well as the variety of configurations and options that are available. The data clearly show that for the same energy output and excluding fuel preparation, suspension burners are the least expensive of the designs. Combustor costs generally increase going from suspension burners to single chamber to multiple chamber, then to fluid-bed designs.

1. Suspension Burners

Suspension burners include units that fire fine materials in a stable flame formed by a burner register and units that fire coarser materials in a refractory-lined furnace chamber, often with cyclonic gas flow. The sub-group of suspension burners that use cyclonic furnace chambers are commonly classified as cyclonic burners. All of these designs can be coupled to water-tube and fire-tube boilers.

Since both pure suspension burners and cyclonic burners rely on combustion of particles in suspension with a carefully controlled amount of air, they have very similar operating characteristics.

An example of a pure suspension burner is one manufactured by Coen Company Ltd. (Figure IV-1). This unit is widely used for installation in water-tube and fire-tube boilers. It burns sanderdust and other pulverized fine wood fuels completely in suspension. Any larger particles not burned completely in suspension fall to complete combustion on a grate or furnace area below.

Typical of the cyclonic burners is the unit manufactured by Energex (Figure IV-2). These units burn larger particles than can be used in a pure suspension burner, utilizing the larger residence time in the hot refractory furnace caused by the cyclonic flow to obtain complete particle burn-out. Combustion air is added at three locations along the furnace, the level of excess air controlling the gas discharge temperature. Excess air levels as low as 15% are possible, but higher levels of 25 - 35% are common.

Cyclonic furnaces can be operated in a slagging or non-slagging mode. Because of severe problems with rapid deterioration of refractory and with removal of liquid ash, non-slagging operation is preferred. Discharge gas temperatures should usually be 1000°C or less to avoid slagging conditions. Problems with slagging are worst for dirty bark due to the high ash content of the bark and the presence of silica in the foreign inerts.

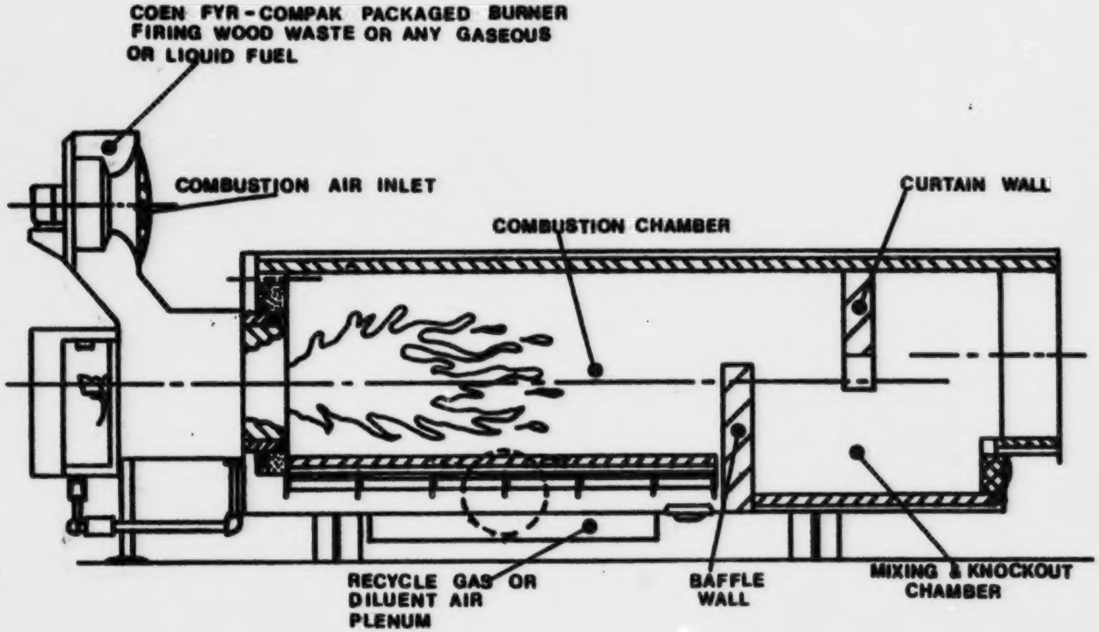

COEN FYR-COMPAK PACKAGED BURNER FIRING WOOD WASTE OR ANY GASEOUS OR LIQUID FUEL
COMBUSTION AIR INLET
COMBUSTION CHAMBER
CURTAIN WALL
RECYCLE GAS OR DILUENT AIR PLENUM
BAFFLE WALL
MIXING & KNOCKOUT CHAMBER

FIGURE IV-1 COEN DAZ SCROLL FEED BURNER

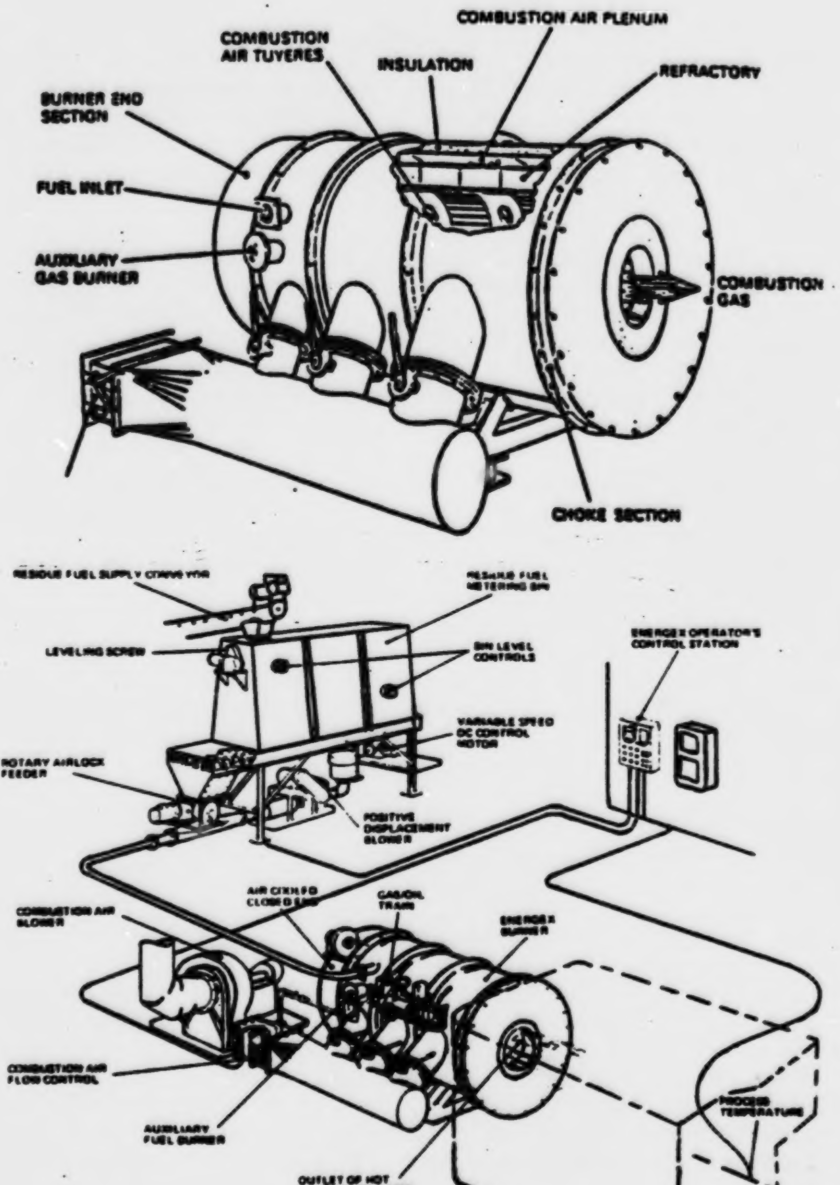


FIGURE IV-2 ENERGEX SUSPENSION BURNER

Most installations of cyclonic burners have been for hot air heating of kilns, veneer dryers, rotary dryers, flash dryers and other dryer types. These burners can also be used for heating boilers, but wall-mounted suspension burners are preferred for this application because of the advantage of radiant heat transfer from the exposed flame to the boiler tubes.

Other designs of suspension burners are available that cannot be classified as pure suspension burners or cyclonic burners, but do rely on combustion of fine wood particles in suspension and use some cyclonic flow. An example of a suspension burner using a primary firing tube and a secondary combustion chamber is the Heat Cell marketed by Georgia Pacific (Figure IV-3).

The performance characteristics of pure suspension burners, cyclonic burners and one other design of suspension burner, the Georgia Pacific Heat Cell, are summarized in Table IV-2. The performance of the suspension and cyclonic burners are generalized because of the similarity in the performance of competing units.

All pure suspension burners need a pilot flame to stabilize combustion of the wood fines. Some burners need pilots sized to 2% of the combustion capacity, while others need pilots sized up to 10% of capacity.

Manufacturers of suspension burners are listed in Table IV-3.

2. Single Chamber Combustors

Single chamber combustors are relatively simple combustors that sustain combustion of wood residues by undergrate and overfire injection of air. Combustion of solids takes place on a grate, while evolved fuel gases are burned above. These units are available in sizes up to about 6 GJ/h, higher capacities being difficult to obtain because of increasing problems with combustion stability and particulate emissions. Most single chamber combustors are used for disposal of residues or generation of hot water or warm air for heating of secondary manufacturing plants with small heating loads.

Because combustion takes place on the fuel pile and in an often poorly stabilized overfire area immediately above the pile, turndown of units in the upper part of the capacity range tends to be limited.

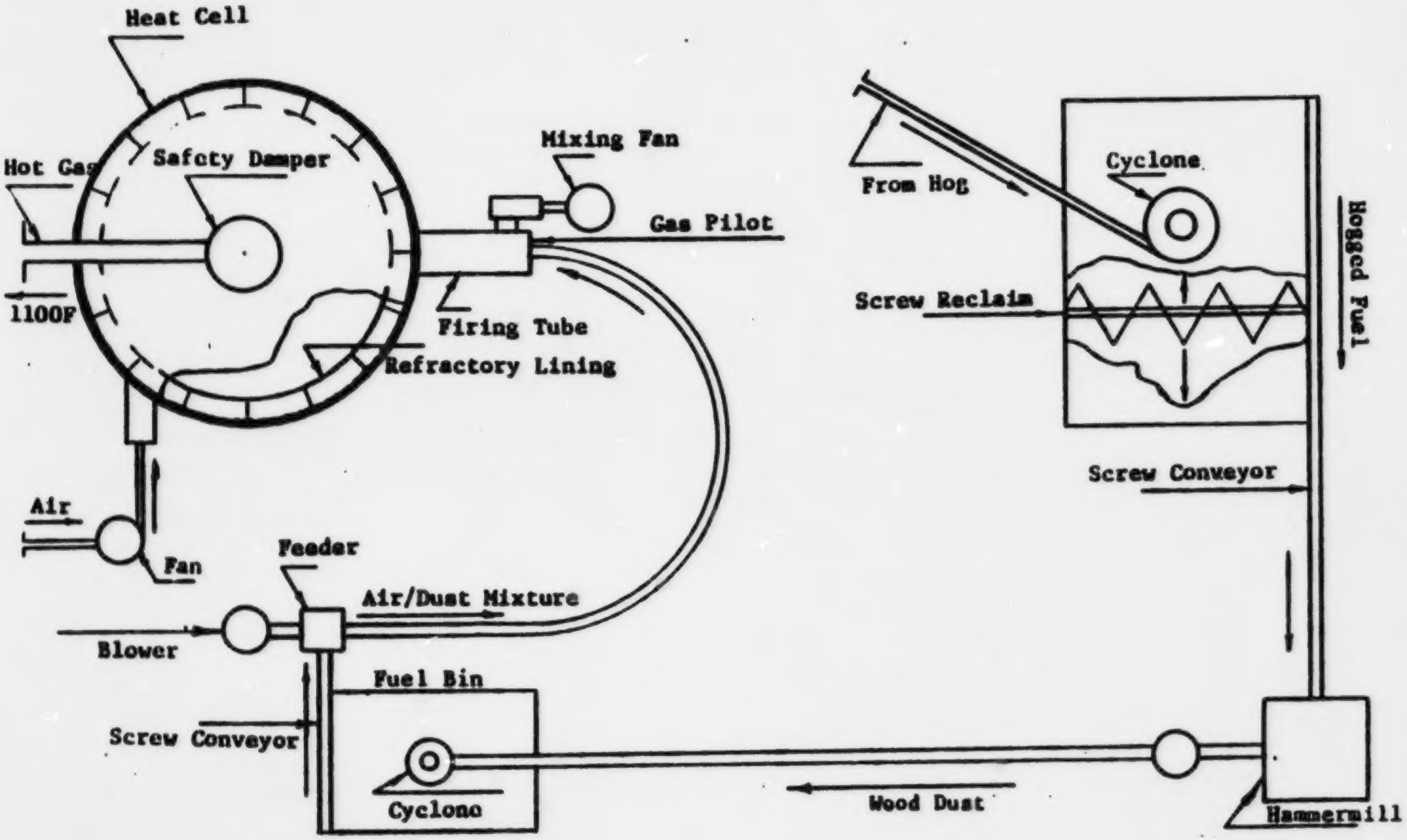


FIGURE IV-3 GEORGIA-PACIFIC HEAT CELL

Table IV-2

PERFORMANCE CHARACTERISTICS OF SUSPENSION BURNERS

Criteria	Suspension Burner Without Furnace Chamber	Suspension Burner With Furnace Chamber	Georgia Pacific Heat Cell
State of Development	Commercial	Commercial	Demonstration
Fuel: Size (cm)	<0.1	<0.3 X 0.3 X 2	<0.3
Moisture Content(% wb)	<10	<15	<30
Output Capacity (GJ/h)	5 - 85	5 - 85	42
Turndown	5 : 1	5 : 1	N/A
Thermal Efficiency (%)[a]	>95	95	
Excess Air (%)	15 - 20	15 - 35[b]	
Hot Gas Temperature[c] (°C)	---	600 - 1200[c]	650
Particulate Loading (g/m^3, 15°C, 12% CO_2)	2.5[d]	2.5[d]	N/A
Power Required For Feeding and Combustion (kWh/GJ)	0.7 - 2	2 - 3.5	
Controls	Automatic	Automatic	Automatic
Flame Pilot	Yes	Yes	Yes
Ease of Operation	Good	Good	N/A
Ease of Maintenance	Good	Good	N/A
Response to Load Change (s)	30 - 60 max.	30 - 60 max.	N/A
Equipment Cost ($/GJ/h)[e]	10 GJ/h-7,800 60 GJ/h-2,000	13,600 3,800	

[a] Allows for unrecoverable losses only - combustibles in ash, ash sensible heat, wall losses

[b] Excess air up to 130% has been used to produce lower gas temperatures

[c] Upper temperature limited by ash slagging behaviour

[d] At burner exit assuming all ash emitted - ash content = 2%,db

[e] Cost estimate includes fuel metering system and combustor

Table IV-3

MANUFACTURERS OF SUSPENSION BURNERS

Burners Only:

Coen Company Ltd.

Peabody Gordon-Piatt

Bumstead-Woolford Co.

Burners With Furnace Chambers:

Coen

Peabody Gordon-Piatt

Energex Ltd. (cyclonic design)

Waycot Systems Ltd.

Guaranty Performance Co. Inc. (cyclonic design)

McConnell Industries (cyclonic design)

Georgia-Pacific Corporation

Lochhead-Haggerty Engineering & Manufacturing Co. Ltd. (cyclonic design)

Pillard Inc. (cyclonic design)

The combustors have a low initial cost because of their simplicity. They also can handle a wide variety of fuels.

Single chamber combustors do not have satisfactory performance for application with electricity generation systems. Their primary negative aspects are: poor turndown, limited output capacity, high emissions and poor combustion stability.

3. Multiple Chamber Combustors

Multiple chamber combustors are a substantial improvement over the simpler single chamber combustors, being proven technology and free from most of the problems common to the single chamber designs.

Multiple chamber combustors are a two-stage combustion device where the infeed is heated and pyrolyzed to produce a char that is then combusted on a grate with a substoichiometric quantity of air, while the released hot combustible pyrolysis gases along with products of char combustion are passed to a secondary furnace and burnt with excess air under stable combustion conditions.

Because air flows are low around the fuel pile on the grate (substoichiometric) and residence time of any entrained fuel particles is increased with use of cyclonic flow in the secondary chamber, particulate emissions from multiple chamber combustors are much lower than for single chamber combustors.

Multiple chamber combustors can burn dry and wet wood. The upper limit on moisture content being upwards of 60%, wet basis. Stable burning of wet wood is aided by preheating of inlet under-fire air, radiative heating from the hot refractory and cyclonic flow of over-fire gases for good mixing of air and combustibles.

An example of a vertically oriented multiple chamber combustor with under-fire feeding is the Wet Cell manufactured by Lamb-Cargate Industries Ltd. (Figure IV-4). In this unit, wood with up to 60% moisture content is gasified on the grate, while the released gases are burnt in the cyclonic flow chamber. Ash, sand and other fine inerts entrained in the gases tend to be swept to the combustor wall and fall to the grate for automatic removal with ash from the fuel pile. Temperatures in the lower chamber are kept below 870°C to increase refractory life and avoid clinkering, while temperatures in the exit gases can reach up to 1300°C because of over-fire combustion. Temperatures are controlled to prevent slagging of ash.

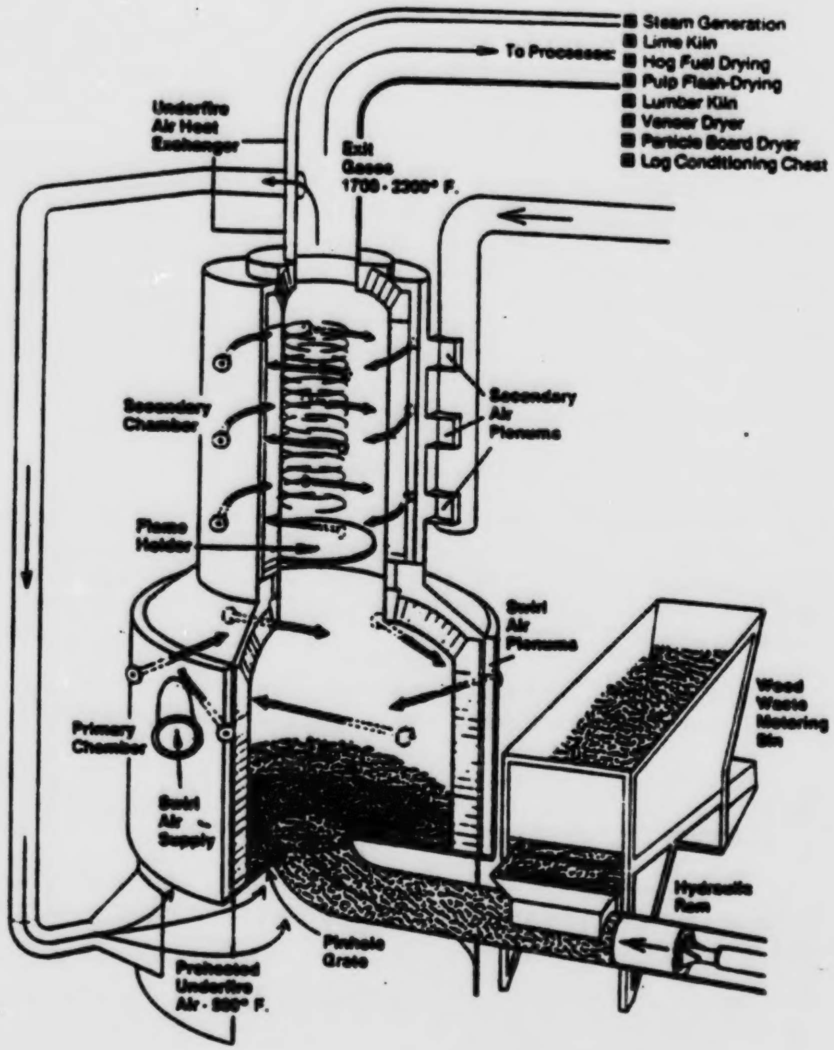


FIGURE IV-4 LAMB-CARGATE WET CELL

Systems are available in sizes from 11 to 160 GJ/h and have turndown ratios of five to one. The response time for a 5:1 load change is claimed to be only 20 seconds. All systems are operated under automatic control.

An example of one of the horizontally oriented multiple chamber combustors is the Anga & Varma Canada Ltd. unit (Figure IV-5); this unit is also available with a boiler above the primary furnace. In this unit dry or wet fuel up to 65% moisture content can be used. Some drying of feed and recycle of unburned carry-over results from the passage of the feed screw through the secondary combustion chamber. Over-fire and under-fire air is used to control combustion in the two combustion chambers. Most systems are complete with a low-pressure boiler (Figure IV-5) with steam pressures below 1,826 kPa and temperatures below 260°C.

Ash removal is semi-automatic, requiring manual raking of ash to one corner of the furnace where it drops to a screw conveyor for removal from the furnace. This is done while the unit is operating to prevent damage of refractory lining by thermal shock.

The furnace typically operates with a gas temperature of 1000°C, although temperatures up to 1100°C are possible.

Systems are available in sizes from 4 to 42 GJ/h. These units are fabricated in the field with reinforced concrete as a permanent installation.

The performance characteristics of two arbitrarily selected, commerically available, multiple chamber combustors are given in Table IV-4.

Manufacturers of multiple chamber combustors are listed in Table IV-5.

4. Fluid-Bed Combustors

Fluid-bed combustors are available as individual units for generation of hot gas, or coupled to a water-tube or fire-tube boiler for steam generation. In these combustors the inlet air for combustion is distributed below the bed using either nozzles, a perforated plate or some similar distributor. The bed is composed of an abrasive-resistant and thermally-stable sand having a particular size distribution conducive to good fluidization. The depth of the sand bed varies with manufacturer of the system and affects the proportion of combustion occurring in the bed and the freeboard

LIST OF ITEMS

1	Hydraulic powerstation
2	Hydraulic cylinders
3	Hydraulic scraper system
4	Screw conveyor outfeed
5	Main feed screw conveyor
6	Stoker bin
7	Stoker screw conveyor
8	Predrying zone for fuel
9	Main furnace
10	Primary air fan
11	Secondary air fan
12	Boiler
13	Fallout zone for unburned particles
14	Dust separation
15	Reintroduction of dust
16	Dust outfeed system
17	Main fan for fumes
18	Smoke stack
19	Outfeed system for ash and cinder
20	Container

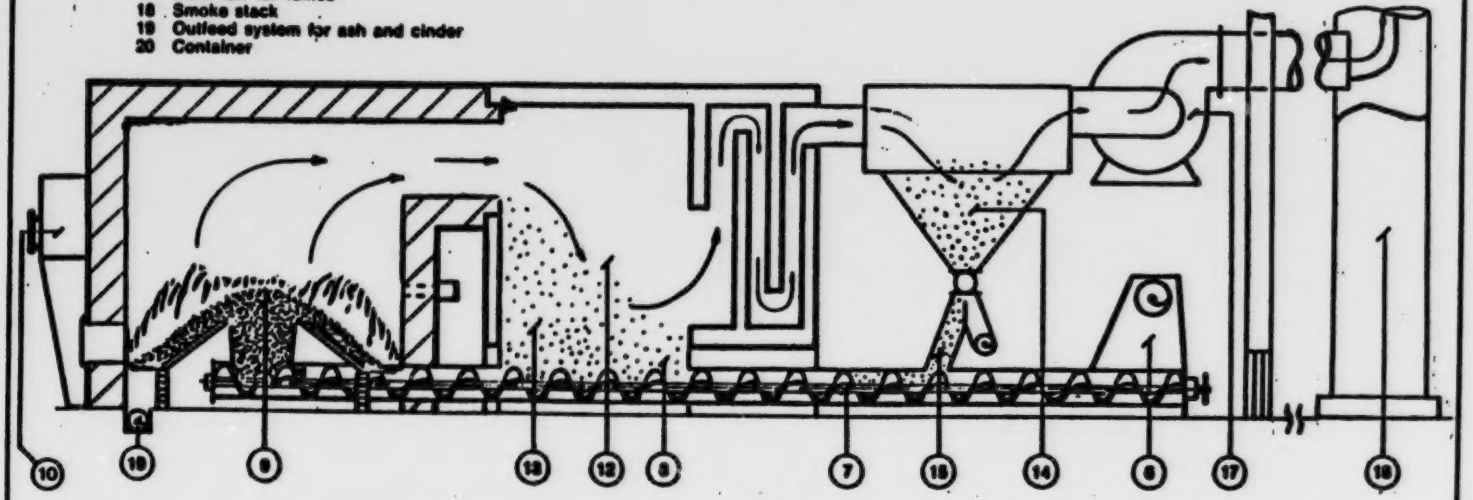


FIGURE IV-5 ÅNGA & VÄRME MULTIPLE CHAMBER COMBUSTOR

Table IV-4

PERFORMANCE CHARACTERISTICS OF MULTIPLE CHAMBER COMBUSTORS

Criteria	Lamb-Cargate Wet Cell	Anga & Varme
State of Development	Commercial	Commercial
Fuel: Size (cm)	<8	<10
Moisture Content(% wb)	<60	<65
Output Capacity (GJ/h)	10-158	4 - 42
Turndown	5 : 1	
Thermal Efficiency (%)[a]	>90	93
Excess Air (%)	15 - 25	25 - 50
Hot Gas Temperature (°C)[b]	870 - 1300	1000 - 1100
Particulate Loading (g/m^3, 15°C, 12% CO_2)	0.14 - 0.37	
Power Required For Feeding and Combustion (kWh/GJ)	7 - 8.5	1 - 3.6[c]
Controls	Automatic	Automatic
Ease of Operation	Good	Good
Ease of Maintenance	Good	Fair
Response to Load Change (s)	20	
Equipment Cost ($/GJ/h)[e]	11 GJ/h-15,000 150 GJ/h-9,200	

[a] Allows for unrecoverable losses only - combustibles in ash, ash sensible heat, wall losses.

[b] At combustor exit

[c] Higher value includes induced draft fan for flue gas.

Table IV-5

MANUFACTURERS OF MULTIPLE CHAMBER COMBUSTORS

Anga & Varme Canada Ltd. (conical grate furnace)

Basic Environmental Engineering Inc. (horizontal furnace)

Berg & Stark A.B. (conical grate furnace)

Konus Systems Inc. (hot oil heaters)

Lamb-Cargate Industries Ltd. (cyclonic furnace)

Witermo Oy (conical and inclined grate furnace) regions. The vigorous bubbling and mixing action of the fluid bed promotes good dispersion of fuel throughout the bed and uniformity of bed temperature. Systems with a deep bed encourage complete combustion of fuel in the bed area and, consequently, bed temperatures are higher than or equal to freeboard temperatures. Conversely, shallow beds promote combustion of large particles in the bed and combustion of volatiles in the freeboard. Freeboard temperatures in this case are usually highest.

An example of a fluid-bed combustor and water-tube boiler combination is illustrated in Figure IV-6. This unit by energy Products of Idaho (EPI) is available in a number of configurations, one being as a combustor only. The combustor can use wood fuels with sizes up to 8 cm and moisture contents up to 65%, wet basis. Reportedly, 40 to 70% of combustion occurs above the 38 to 61 cm deep sand bed.

Hot gas at 930 - 1100°C with some entrained ash and fine bed material is discharged from the combustor and passed directly into a downstream boiler, blend box or heat exchanger. Flue gas is cleaned with cyclones or multiple cyclones as needed before discharge to the atmosphere.

Units are available with capacities up to 127 GJ/h and combustor diameters of 2.3 to 6.7 m. Turndowns are limited by the fluidization characteristics of the bed; lower capacities are limited by defluidization of the bed, while upper capacities by carryover and heat release. A freeboard region is needed to allow heavier particles that may be carried out of the bed to disengage and fall back to the bed.

An installation at Maibec Industries, St. Pamphite, Quebec, has a capacity of 14 GJ/h, diameter of 2.3 m ($3.4\ GJ/hm^2$), height of 8.2 m and fluidization air inlet pressure of 14 kPa (gauge). Temperatures run near 800°C in the bed and 1060°C in the freeboard.

An installation at Kirby Forest Industries, Silsbee, Texas, has a normal capacity of 106 GJ/h and heat release of $4.2\ GJ/hm^2$.

EPI uses 5 cm of insulating block under 10 cm of castable refractory. Refractory life is from 6 to 10 years, depending on operating conditions, especially the variability of operating temperatures.

Most fluid-bed combustors are used for heating, drying, and firing of water-tube and fire-tube boilers. Boiler installations are typically for steam at 1,140 to 1,830 kPa.

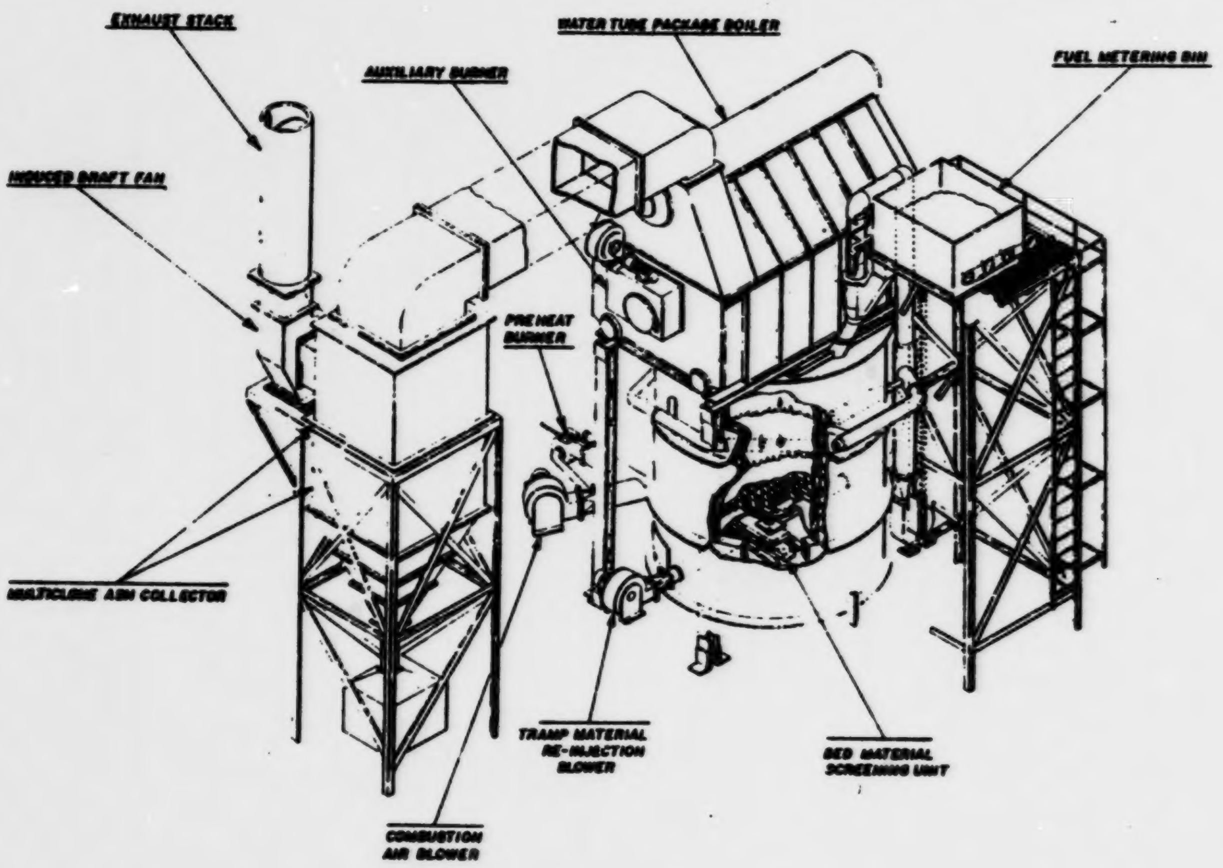


FIGURE IV-6 ENERGY PRODUCTS OF IDAHO FLUID-BED COMBUSTOR

Andrew of Boise Cascade Corporaton observed particulate loading of up to 4.6 g/m^3 (15°C, dry) at the exit from an EPI fluid-bed combustor burning pine bark. Use of multiclones reduced this to 0.58 - 0.7 g/m^3 for use in a dryer. The uncleaned gas was directed into a fire-tube boiler (14 kg/h steam) and caused problems by plugging of tubes, erosion at tube bends and high wear of an induced-draft fan. The tubes needed to be cleaned every six months. Andrew also indicates fluid-bed combustors to supply heat to dryers and a firetube boiler will have a power requirement of about 4.8 kWh/GJ. One unit of 169 GJ/h had a designed power requirement, excluding fuel preparation, of about 5 kWh/GJ for air and fuel injection, and ash removal.

A summary of performance data for two manufacturers of fluid-bed combustors is presented in Table IV-6.

Table IV-7 lists manufacturers of fluid-bed combustors. Companies marketing integral fluid-bed combustors/boilers are dealt with in the section on boilers.

D. Gasifiers

Gasifiers can be classified according to the state of motion of the fuel bed - fixed (packed), stirred or fluid - and, for fixed-bed units, according to the relative direction of motion of the fuel bed and injected gas reagents - updraft or downdraft. Updraft, downdraft, fluid-bed and stirred bed gasifiers generate a fuel gas with similar heating value, but have substantially different fuel requirements, operating performance characteristics and, of particular importance, gas compositions and yield of condensible hydrocarbons.

The simpler of the two common designs of fixed-bed gasifiers is the updraft design (Figure IV-7). In this gasifier, air or some other reagent is introduced at the base of a fuel bed while fuel is replenished from the top. Fuel gas is withdrawn from the top of the fuel bed after passing by the incoming feed and, consequently, contains pyrolysis products, including tars, oils and water vapour.

In a downdraft design (Figure IV-8) the positions of the combustion and reduction zones have been interchanged compared to the updraft design. Some gasifier designs pass the fuel gas over the outside of the section of the reactor containing the pyrolysis and drying zones to improve thermal efficiency. Since the fuel gas passes through a porous hot bed of char in the reduction zone, it contains less than 10% of the condensible hydrocarbons produced from an updraft design. The fuel gas exits at a high temperature and with a low enough velocity to let most ash produced in the combustion and reduction zones settle out.

Table IV-6

PERFORMANCE CHARACTERISTICS OF FLUID-BED COMBUSTORS

Criteria	Energy Products of Idaho	Thermal Processes Inc.
State of Development	Commercial	Demonstrated
Fuel: Size (cm)	<8	<15
Moisture Content (% wb)	<60	<65
Output Capacity (GJ/h)	11 - 127	11 - 130
Turndown	2 - 3 : 1	2 - 3 : 1
Thermal Efficiency (%)[a]	95	
Excess Air (%)	50 - 100	85 - 100
Hot Gas Temperature (°C)	930 - 1200°C	1000°C
Particulate Loading [b] (g/m^3, 15°C, 12% CO_2)	<.03-.015	
Power Required For Feeding and Combustion (kWh/GJ)	2.5-5	3.0 - 3.2
Controls	Automatic	
Ease of Operation	Good	Good
Ease of Maintenance	Good	Fair
Response to Load Change (s)	---	---
Equipment Cost ($/GJ/h)	12.7 GJ/h-29,500 176 GJ/h-5,300	11 GJ/h-37,000 100 GJ/h-8,500

---

[a] Allows for unrecoverable losses only - combustibles in ash, ash sensible heat, wall losses

[b] At combustor exit

Table IV-7

MANUFACTURERS OF FLUID-BED COMBUSTORS

Energy Products of Idaho Inc.
Combustion Power Co. Inc.
Dorr-Oliver Canada Ltd.
Thermal Processes Inc.
Applied Combustion Systems Ltd.
Copeland Systems Inc.

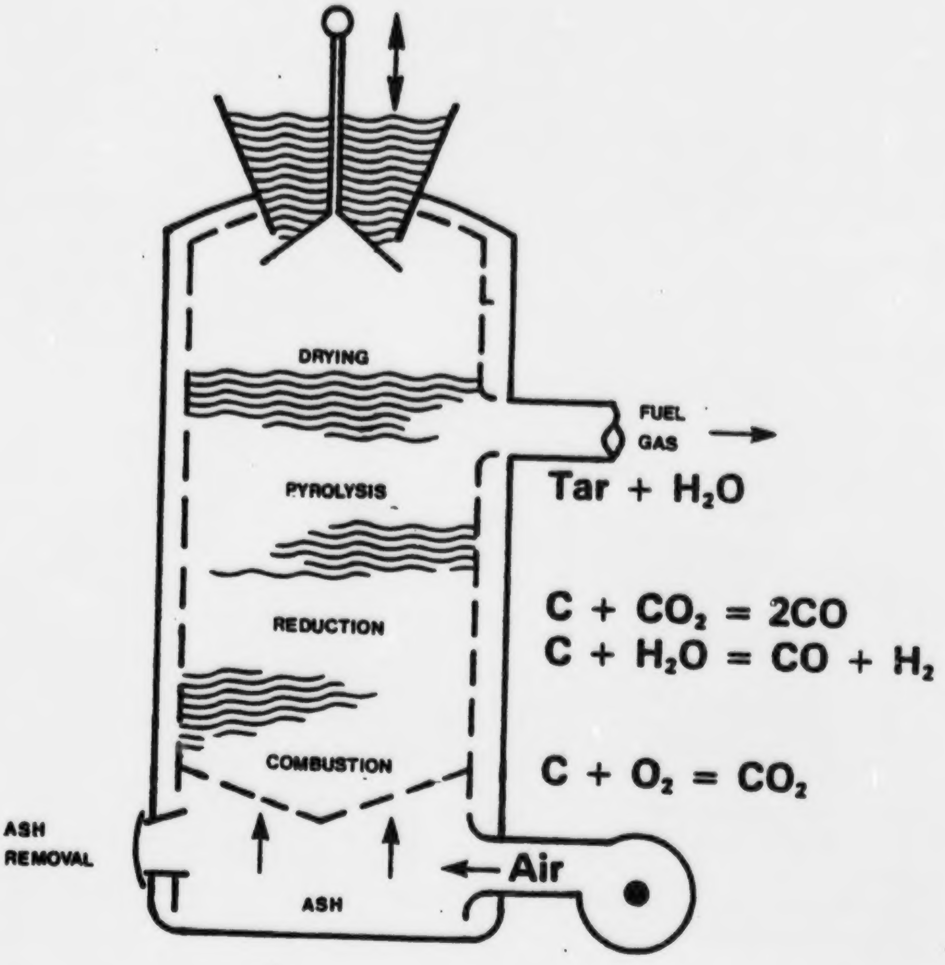
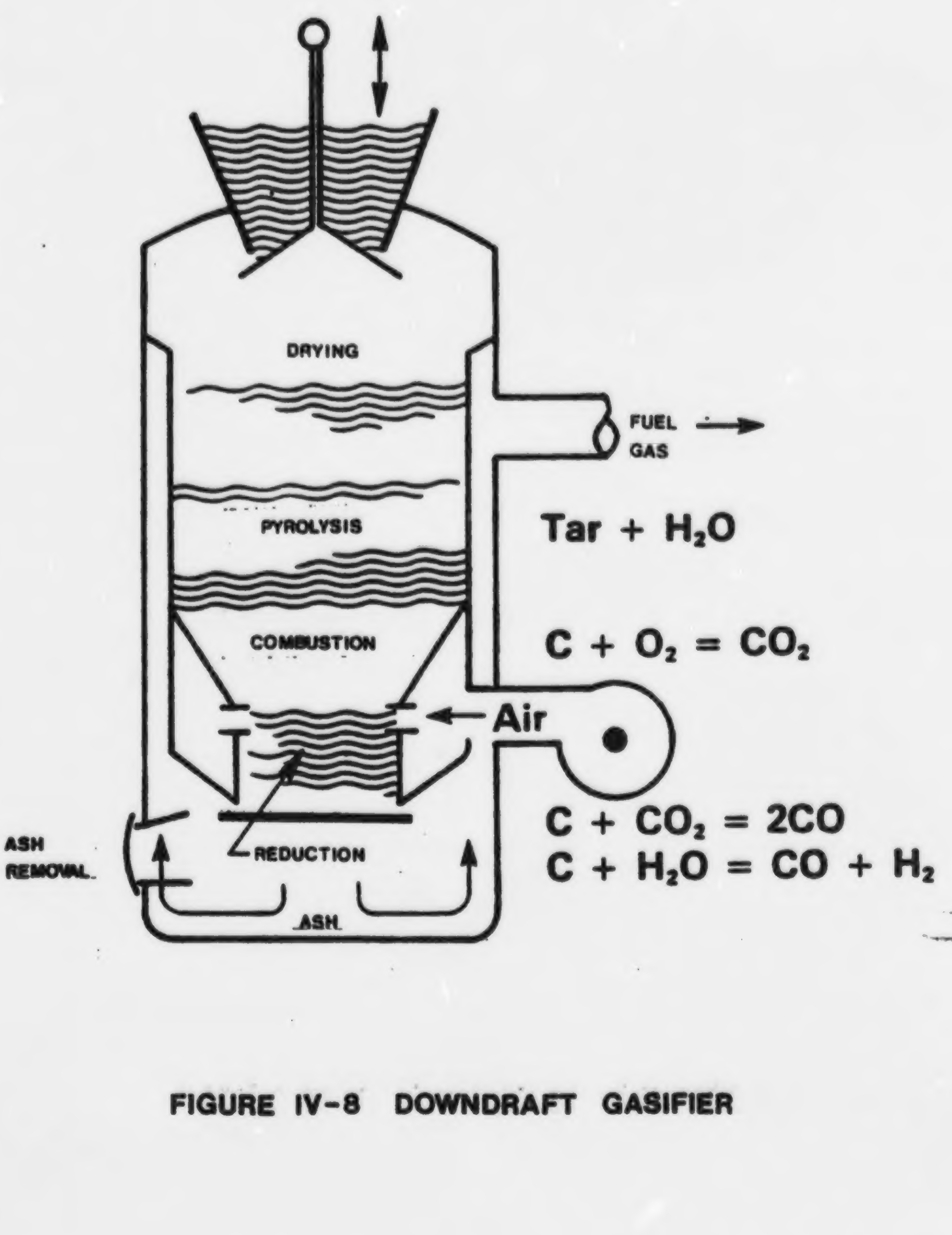


FIGURE IV-7 UPDRAFT GASIFIER



$Tar + H_2O$

$C + O_2 = CO_2$

$C + CO_2 = 2CO$

$C + H_2O = CO + H_2$

FIGURE IV-8 DOWNDRAFT GASIFIER

Updraft and downdraft designs require particle sizes from roughly 0.6 cm to, preferably, 6 cm to produce a fuel bed that is permeable for flow of gases. Downdraft gasifiers require waste fuels with moisture contents in the range of 0 to 30%, wet basis while updraft gasifiers can use fuels with moisture contents of 0 to 50%, wet basis.

In a fluid-bed gasifier (Figure IV-9), air or some other reagent is injected through a distributor to fluidize a bed of granular inert material along with a small proportion (<5% by weight) of fuel particles. Commercial gasifiers are available with either over-bed feeding or in-bed feeding. In either case, the vigorous mixing action of the hot fluidized bed is sufficient to rapidly distribute the feed through the bed volume.

Contrary to the fixed-bed designs, no distinct drying, pyrolysis combustion and reduction zones are present in the bed, although more combustion tends to occur near the distributor where the oxygen concentration is greatest. The conversion processes occur throughout the fluid bed and, in the cases of the drying and pyrolysis processes, at a very rapid rate because of the high temperature of the bed and the excellant heat transfer rate that is possible between the hot solids and gases in contact with the fuel particles.

Fluid beds have higher carryover of solids than fixed beds because of the higher gas velocities needed to fluidize the bed solids. This carryover consists of fuel ash and fine solids produced by deterioration of the bed at high temperatures and under intense bubbling and mixing conditions.

To assist in the preliminary selection of a gasifier appropriate to a system configuration for power generation, a brief summary of performance data for updraft, downdraft and fluid-bed designs is presented in Table IV-8.

The reagent most commonly used by near commercial or commercial equipment for gasification is air. Air gasification produces a low heating value gas with a higher heating value of 4.4 to 6.5 MJ/m^3 (dry gas). Gasification systems may use oxygen or a mixture of steam and oxygen to generate gas with a medium heating value (9 to 13 MJ/m^3, dry gas). More advanced gasification processes have been investigated that gasify in hydrogen atmospheres, or with external heating and a catalyst.

1. Updraft Gasifiers

Updraft gasifiers are simpler in design than the alternate downdraft gasifiers since air or other reagents must only be distributed across the bottom of the reaction chamber. The injected air flows counter currently with the bed materials, leading to high temperatures (1100°C) at the bottom of the bed in the combustion zone, and low temperatures (75 to 100°C) at the top of the bed in

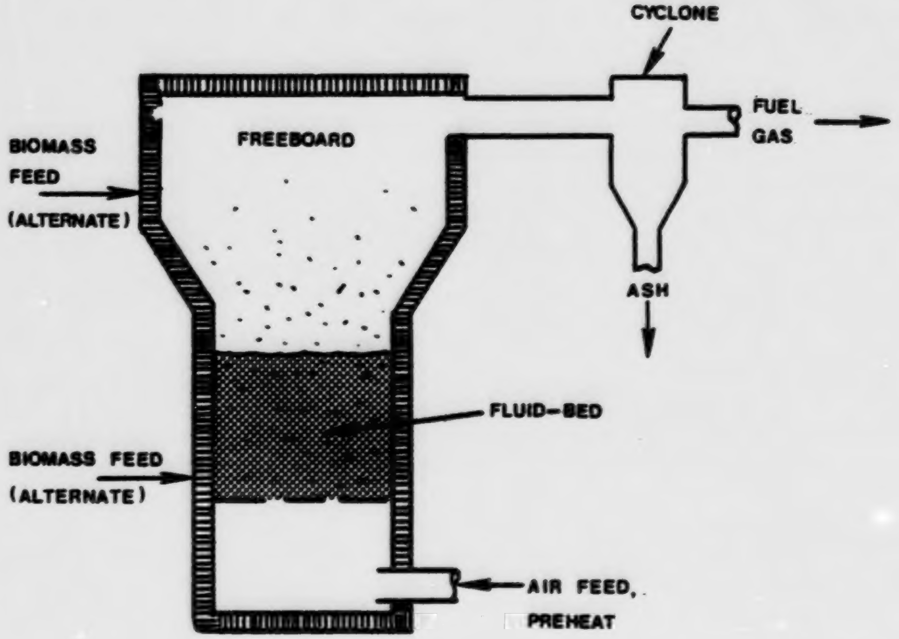


FIGURE IV-9 FLUID-BED GASIFIER

Table IV-8

TYPICAL OPERATING PERFORMANCE AND COSTS FOR GASIFIERS

CRITERIA	UPDRAFT	DOWNDRAFT	FLUID-BED
State of Development	Demonstration	Commercial	Demonstration
Fuel: Size (cm)	0.6 - 5	1 - 10	<5
Moisture Content (%,wb)	25 - 50	<30	<50
Output Capacity (GJ/h)[a]	up to 30	up to 15	up to 150
Turndown	3 - 4 : 1	3 : 1	2.5 : 1
Cold Gas Conversion Efficiency (%)[a]	70 - 75	70 - 85	70 - 80
Air/Fuel Ratio (kg/kg)	1.5 - 2.0	1 - 2	2 - 3
Gas HHV (MJ/m^3)	4.4 - 6.5	4.4 - 6.5	4.4 - 6.5
Exit Gas Temperature (°C)	80 - 95	350 - 600	760
Condensibles:			
Yield			
H_2O (kg/kg wet feed)	0.4	N/A	0.3 - 0.5
Tar (kg/kg dry feed)	0.05	very low	0.02
Tar HHV (MJ/kg)	27.9	N/A	23.3
Particle Loading (g/m^3, 15°C, 101 kPa)	N/A	N/A	N/A
Equipment Cost ($/GJ/h)	25 GJ/h - (U.S.) 30,000	3 GJ/h - (U.S.) 52,000 13 GJ/h - (U.S.) 25,000	63 GJ/h- 13,000

[a] Based on higher heating value of fuel gas at 101 kPa and 15°C.

the drying zone. Condensible tars and oils evolved in the pyrolysis zone are swept along with the gas flow into cooler regions, depositing in the bed and on the upper reactor walls as well as being carried out with the fuel gas. These mists and vapours can be burned with the fuel gas if the gas is kept hot or may be removed from the gas by cooling and cleaning equipment.

Temperatures in the combustion zone must be regulated to prevent slagging and clinkering of the ash. In commercial and development updraft gasifiers, temperature control has been done by regulation of air flows and/or by injection of steam with air or oxygen. Limiting heat release by control of air flow for a given grate area has the obvious disadvantage of limiting the capacity of the gasifier. Steam injection is a preferable means of temperature control, since the steam can react endothermically with char in the bed to produce hydrogen and carbon monoxide.

Updraft gasifiers can tolerate a much higher feed moisture content than comparably sized downdraft designs. Injection of water with feed carries a penalty since the water is driven off as steam at the top of the bed and limits the temperature of this region to below the saturation temperature.

The evolved steam promotes cracking of the heavy hydrocarbons evolved from the pyrolysis zone, but cannot participate in the steam-char reaction, one of the two most important reactions for production of fuel gas (Figure IV-7). Since drying of the feed particles is time-dependent, some drying will occur simultaneously with pyrolysis and, possibly, combustion, although this latter situation is unlikely. Steam for the steam-char reaction in the reduction zone comes primarily from oxidation of the hydrogen present in the wood (for the case where steam is not injected). Extensive testing has been conducted with updraft coal gasifiers, these being of identical configuration to some wood gasifiers, to determine the optimum steam/fuel ratio for gasification. Data defining optimal operation of updraft wood gasifiers does not presently exist.

The disadvantages of using wet fuel in an updraft gasifier can be summarized as follows:

- The evaporation of water within the reactor adds to the heat that must be supplied by the combustion process to maintain a designed bed temperature, thus decreasing the efficiency of conversion to cold fuel gas. Drying of fuel outside the gasifier will increase the efficiency of the gasifier, although this would result in a system with a poorer overall efficiency than for a gasifier alone because of higher system (dryer and gasifier) heat losses. The heat for evaporation is essentially unrecoverable.

- The evolved water vapour dilutes the fuel gas and lowers its as-produced heating value.

- If the fuel gas is cooled to removed condensibles, the resulting condensate will consist of water contaminated with phenolic compounds. This material is a disposal problem and may need to be incinerated.

Updraft gasifiers with air injection produce a gas with a higher heating value of about 5.6 MJ/m^3 (101 kPa, 15°C) when clean. When tar vapours are present in the gas as produced, heating values can reach upwards of 7.4 MJ/m^3. Since the steam-char reaction plays a minor role in updraft gasifiers and exit gas temperatures are too low for water-gas shift reactions to be important, the evolved gas contains a much higher volume fraction of carbon monoxide than hydrogen, typically being in a ratio of about 2 : 1 when using wet wood fuel.

A survey of commercial gasifiers shows that the grate energy release rate for updraft gasifiers falls commonly in the range of 2.5 to 5.0 GJ/hm^2, based on clean, cold fuel gas output. This applies to gasifiers using wood of 40 to 50% moisture content, wet basis.

The air/fuel ratio is important since it strongly influences the amount of feed that is burned to supply energy for gasification. On a weight basis, air/fuel ratios of commercial equipment are usually near 1.5 to 2.0.

Gasifier energy conversion efficiencies can be presented in numerous ways, depending on whether the gas is clean or dirty, or hot or cold. The efficiency of conversion to clean, cold gas is 70 to 75%. Conversion efficiencies to hot dirty gas are higher because of sensible heat and the heating value of tars. These efficiences may be in the range of 80 to 90%.

An example of a commercial updraft gasifier is the Applied Engineering Co. unit shown in Figure IV-10. The version shown is installed at the Suwanee River Plant of the Florida Power Corporation. A slightly different, earlier version began operating at the Northwest Georgia Regional Hospital in Rome, Georgia in 1981. This unit has an output capacity of about 25 GJ/h with 5 cm X 5 cm whole tree chips having a moisture content of 40 to 50%, wet basis, and a heating value of 18.6 MJ/kg, dry basis. The hospital uses the gas in a 8,600 kg/h low pressure boiler. Tars and oils are burned with the fuel gas in a dual fuel burner.

The latest gasifier design incorporates several unique features. The bottom portion of the reactor shell is water-cooled, and acts as a feed water heater for the boiler system. The ash removal system is an improvement over the earlier design since it

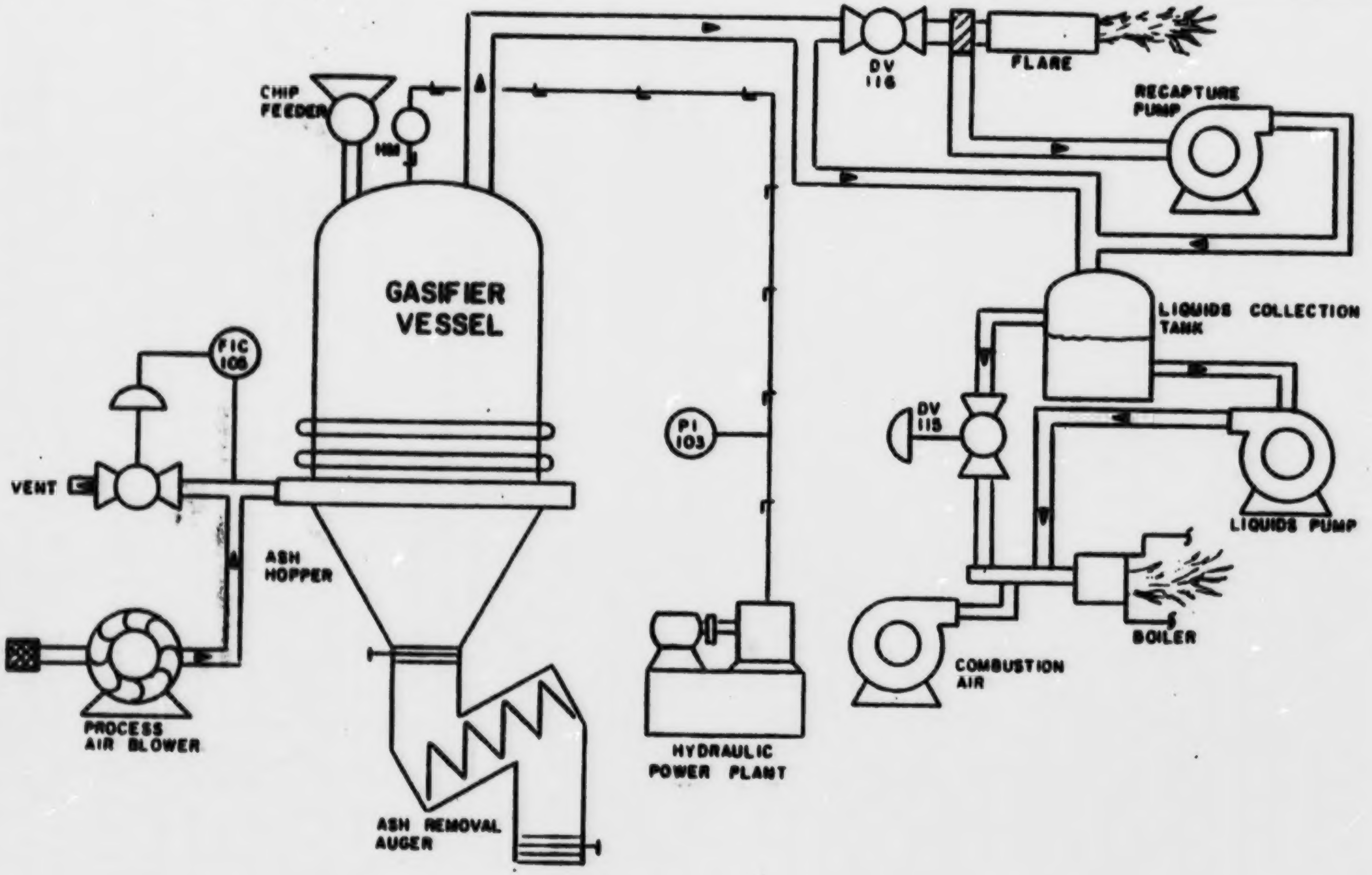


FIGURE IV-10 APPLIED ENGINEERING CO. GASIFIER

provides a means of sealing the ash hopper while still allowing for semi-automatic ash removal. The grate is a proprietary design made of parallel rocking bars that prevent plugging. A bin activator is used in the ash hopper to make the ash flow to the removal screw.

A summary of the performance characteristics of systems marketed by Applied Engineering Co. and Westwood Polygas is presented in Table IV-9.

Very important to the use of fuel gas for electricity generation is the amount of condensibles in the gas as discharged from the reactor and after cleanup. Very little data concerning the yield of tar and water from updraft gasifiers has been documented. Besides the data shown in Table IV-9, D. M. International Inc. has estimated tar yields of 0.05 kg/kg dry wood and condensed water yields of 0.69 kg/kg wet wood for gasification (updraft design) of mixed hardwoods having a size of 1.25 to 4 cm and a moisture content of 50%, wet basis. These figures are similar to data shown in Table IV-9.

No conclusive information on the maintenance requirements and operating behavior of updraft gasifiers over a long term are presently available since commercial units have only recently been installed. Because of the moderate temperatures of the bed near the wall of an updraft gasifier, refractory maintenance should be low. The high temperatures present in the vicinity of the grate and ash removal equipment present some problems for materials selection, although stainless steel or cooled grates have both been used successfully.

The main problem encountered with updraft gasifiers to date has been bridging and irregular downward flow of solids in the bed. This promotes channelling, bypassing of air and generally poorer gasifier performance. As in coal gasifiers, some type of bed agitator is recommended to both promote formation of a bed of uniform thickness and prevent bridging in the bed.

One common problem experienced by updraft gasifiers is tar deposition on upper wall surfaces, the exposed parts of the feed device and the pile-height sensor. Capacitance-type pile-height sensors have suffered from operating problems because of changes in sensitivity with increasing tar deposition. Pile-height sensors that are not exposed to the fuel gas are recommended. Slide gate feed valves have been a problem after shutdown or in cold weather because of solidification of the tars and sticking of the slide gate.

Table IV-10 lists manufacturers of updraft gasifiers and the size, blast composition, number of operating units and status of development for the units that are available.

Table IV-9

PERFORMANCE CHARACTERISTICS OF UPDRAFT GASIFIERS

CRITERIA	APPLIED ENGINEERING CO.	WESTWOOD POLYGAS
State of Development	Demonstration/ Commercial	pilot
Fuel: Size (cm)	5 X 5 chips	<10
Moisture Content (%,wb)	40 - 50	25 - 50
Blast Composition	Air	Air
Output Capacity[a] (GJ/h)	25	30
Turndown	3 - 4 : 1	4 : 1
Cold Gas Efficiency[a] (%)		75
Exit Gas Temperature (°C)	80 - 95	80 - 95
Fuel Gas:		
Non-Condensibles		
Yield (Vol. %, dry)		
N_2	52	46
CO	28	27
CO_2	7	9
H_2	11	16
CH_4	2	2
C_{2+}		N/A
HHV (MJ/m^3)	5.4	5.4
Condensibles		
Yield		
H_2O (kg/kg wet feed)		0.43[b]
Tar (kg/kg dry feed)	0.05 - 0.06	
Tar HHV (MJ/kg)	27.9	
Particulate[c] g/m^3 (101 kPa,15°C)		0.06 (est.)
Power Required For Feeding and Gasification (kWh/GJ)		
Controls	Automatic	Automatic
Ease of Operation	Fair	Unknown
Response to Load Change		
Equipment Cost ($/GJ/h)	25 GJ/h - (U.S.) 30,000	

[a] Based on higher heating value of fuel gas at 101 kPa and 15°C

[b] Contains tar removed in scrubber at 40°C

[c] After gas cleanup

Table IV-10

MANUFACTURERS OF UPDRAFT GASIFIERS

MANUFACTURER	BLAST COMPOSITION	OUTPUT CAPACITY (GJ/h)	OPERATING UNITS	STATUS OF DEVELOPMENT
Applied Engineering Co.	Air	25	2	Demonstration
Century Research Inc.	Air	25 - 50	None	Pilot
D.M. International Inc.	Air	40 - 50	20 (originally)	Demonstration/ Commercial
Halcyon Assoc. Inc.	Air	25 - 50	1	Demonstration
Westwood Polygas Ltd.	Air & Steam	20 - 30	1	Pilot

2. Downdraft Gasifiers

The configuration of the air distribution system for a downdraft gasifier is more elaborate than what is needed for an updraft design. Solids are injected at the top of the bed and ash is withdrawn at its base. Air is injected at several locations around a flow constriction (throat) placed a sufficient distance above the grate to allow formation of a good bed of char for a reduction zone. High temperatures (1100°C) are achieved in the combustion zone produced at the throat. These combustion products pass through the reduction zone where CO_2 is reduced to CO. All the pyrolysis gas and steam evolved above the throat passes through the porous, hot combustion and reduction zones resulting in cracking of the heavier pyrolysis hydrocarbons and gasification of char with steam. Ash is removed from the base of the bed by movement of the grate.

Commercial downdraft gasifiers route the fuel gas up along a shell containing the bed to recover some thermal energy and supply heat needed in the pyrolysis and drying zones.

Temperatures in the combustion zone need to be kept below the ash softening point to prevent clinkering of ash, and solids flow problems. The common practice for temperature control is by regulating the air flow rate. The maximum output capacity is therefore dictated by the rate of char combustion and reduction in and below the throat area.

Downdraft gasifiers cannot tolerate as high a feed moisture content as the updraft designs. This is a consequence of the configuration of the bed which requires that all produced steam flow through the bed. Heating of the upper areas of the bed occurs solely by conduction, which is much less effective than heat transfer by convection as occurs with the updraft design. An increase in the moisture content of the feed results in additional heat requirements for evaporation and additional consumption of feed to achieve the required temperature in the combustion and reduction zones, lowering conversion efficiency.

Most manufacturers of downdraft gasifiers recommend feed moisture contents less than 30%, wet basis and prefer moisture contents less than 20%, wet basis.

Steam evolved from moisture in the wood and formed from the hydrogen constituent of the wood participates in steam/char reactions. The fraction of the evolved steam participating in the gasification reactions, or, alternatively, the optimum steam/wood ratio, have not been documented in the public literature by any of the manufacturers of downdraft gasifiers. Graham and Huffman have reported some data for tests with a Biomass Corp. gasifier. These tests showed that an increase in feed moisture from 3.6%, wet basis, to 26.1% led to a decrease in cold gas efficiency from 64% to 28% and a corresponding increase in the required air/fuel ratio from 1.2 to 1.6 kg/kg. Because of the lack of data, optimum feed moisture contents for a commercial installation cannot as yet be specified and would have to be determined as needed.

Downdraft gasifiers require somewhat more preparation of the size distribution of the feed than do updraft gasifiers, although no manufacturers have detailed specifications for fuel preparation. Preferably, the feed should consist of pieces of the size of chips and small blocks, and have only a small fraction of fines of -1 cm (Graham and Huffman). Fine particles produce a fuel bed with a pressure drop that is too high for satisfactory gasifier operation.

Downdraft gasifiers with air injection produce a gas with a higher heating value of 4.5 to 7.0 MJ/m^3, 5.0 MJ/m^3 being a typical average. The produced gas contains very small amounts of condensible tars because of the passage of tars through the hot combustion and reduction zones. Tests conducted by Van Der Heijden et al with an Imbert gasifier (output capacity 0.9 GJ/h) have shown that gas heating value is not very sensitive to operating load, although the gas composition varies considerably. The gas heating value is strongly affected by fuel moisture content. Tests with a Biomass Corp. gasifier (output capacity at 0.8 GJ/h) have indicated that gas heating values decrease from 7.1 MJ/m^3 to 5.0 MJ/m^3 when fuel moisture content was increased from 13% to 34%, wet basis.

Data for compositions of fuel gas produced by Imbert and Biomass Corp. gasifiers show that the CO/H_2 mole ratio is about 0.8 to 1.1. On average, this is about one-half of the value for updraft gasifiers and indicates the importance of the steam-char and tar cracking reactions to performance of the downdraft design.

A survey of the relatively small downdraft gasifiers available commercially showed that the grate energy release rate is commonly between 4.5 and 6.5 GJ/hm^2, based on output of clean, cold fuel gas. The average of this range is 45% higher than the average of the range (2.5 to 5 GJ/hm^2) found for updraft gasifiers. Higher energy release rates are possible with downdraft designs because the structure of the fuel bed reduces the problem with entrainment of solids with the fuel gas (the ash is held by the grate), and that the pressure gradient in the bed acts in the same direction as gravity, promoting downward motion of the bed solids.

The air/fuel ratio cannot be easily varied for most commercial gasifiers because of inherent performance characteristics of the design. Tests conducted with a Biomass Corp. gasifier (L'Ecuyer and Huffman) observed air/fuel ratios of 1.34 to 1.72 for whole-tree hybrid poplar chips with moisture contents of 13 and 34%, respectively. The added heat load of the wetter chips leads to the higher observed air/fuel ratios. The observed air/fuel ratios are very similar to those used for updraft gasifiers.

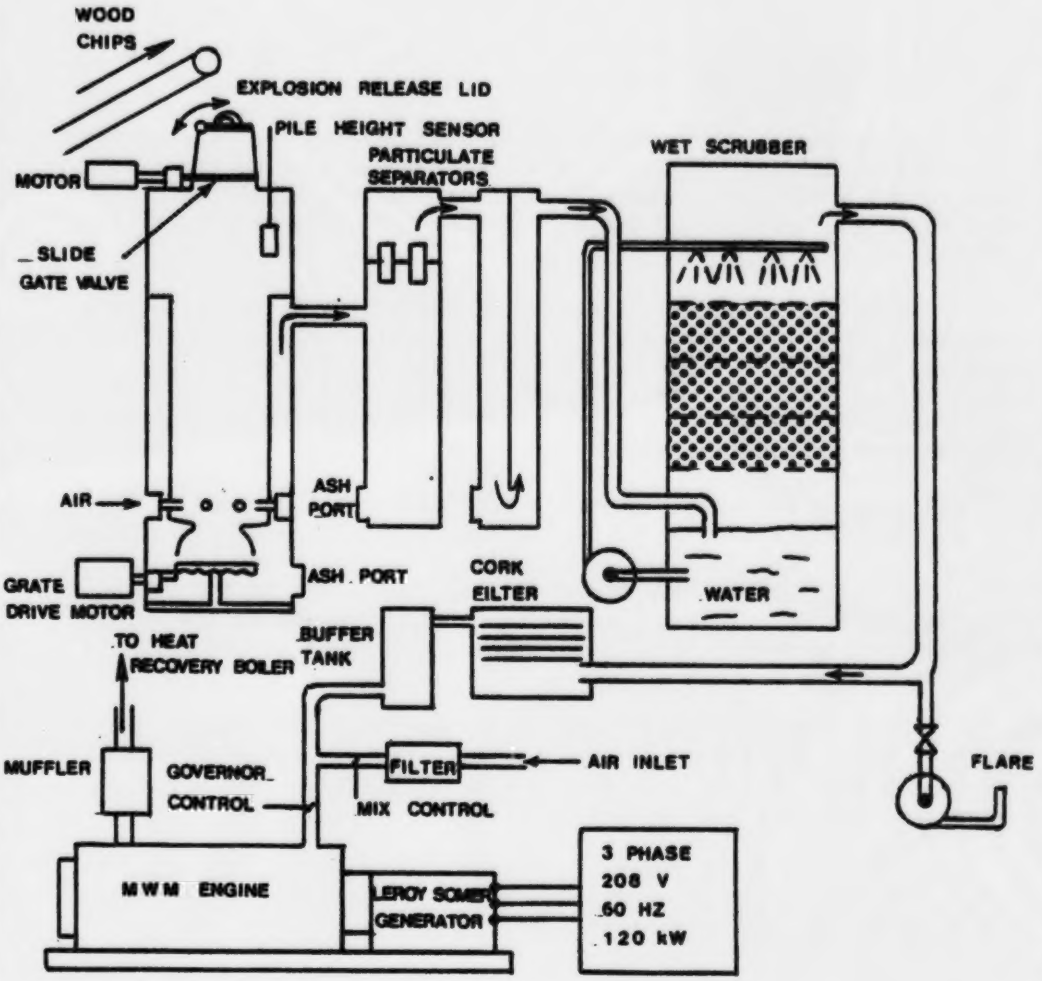
Gasifier energy conversion efficiencies can be presented on a hot-gas or cold-gas basis. Little change of heating value occurs with scrubbing of the gas because of its low tar and oil content. Conversion efficiencies based on cold gas heating value were found to be between 72 and 79% for a small Imbert gasifier using wood chips with a moisture content of 9 to 12% (Van Der Heijden et al). The efficiencies of a Biomass Corp. gasifier using whole-tree chips (13 to 34% moisture content) have been determined to be 74 - 55% (L'Ecuyer and Huffman). Hot gas efficiences, of course, may exceed 85%, depending on heat losses from the reactor structure.

An example of a downdraft gasifier installation is the Imbert unit operated by Snoeshoe Industries of Fort Providence, N.W.T. This gasifier (Figure IV-11) is connected to gas cleanup equipment and a water-cooled MWM spark ignition diesel engine and attached generator. The system is nominally rated at 120 kW output, putting the gasifier output at about 2 GJ/h (energy input of near 2.7 GJ/h). The gasifier is equipped with an on/off automatic feed control activated by an internally mounted capacitance-type pile-height sensor. Sealing of the gasifier is achieved by use of a lock-hopper and inlet and outlet valves.

The system was installed in 1981 and commissioned in the fall of that year. Numerous relatively minor problems were encountered. The most serious operating problems to date have been with the automatic feed system which was designed by Imbert:

- Pyrolysis fumes are discharged from the top hatch of the lock hopper at the start of the feed cycle.
- Wood spills over the top of the reactor when feeding.
- Sensing of pile height is not dependable due to changes in the sensitivity of the pile-height sensor as it becomes covered with tar.
- bridging prevents proper filling of the apron conveyor and, since this conveyor is on a cycle-time control, this leads to irregular feed rates.
- Some problems have also been encountered with bridging within the reactor.

A Koppo chipper is used to chip the feed, presently Jackpine recovered from a burnt area and having little bark. The chips are dried by under-floor heating of a fuel pile, resulting in chips with an as-fed moisture content of about 4%, wet basis. The common



PLAN AREA INCLUDING FEED HOPPER: 50 m^2 (4.5 X 11 m)
MAXIMUM HEIGHT: 9 m

FIGURE IV-11 IMBERT GASIFIER INSTALLATION AT SNOWSHOE INDUSTRIES, FORT PROVIDENCE, NWT

size of the chips is 1.3 to 2 cm thick by 6.4 X 6.4 cm, although butt ends of the 10 cm diameter logs are present as well.

The system is very responsive to load changes and is also easy to start, requiring only a few minutes from ignition of the char bed to gas production. Variations in gas quality appear to cause the observed variation of the electricity frequency between 60 and 61 Hz.

Ash is removed from the bottom of the bed by a rotating grate operated periodically on a cycle-time control. This ash must be removed manually from the bottom of the gasifier through a port when the unit is shut down. Imbert suggests this be done weekly, although the size of the ash hopper dictates more frequent cleaning if the unit is run continuously.

A summary of performance data obtained by Van Der Heijden et al during tests with a 52 kW mobile Imbert gasifier and air-cooled diesel (spark ignition) set fired with whole tree chips is presented in Table IV-11. Of particular importance to the characteristics of the complete electricity generating set is the rapid decline in the overall efficiency when the load falls below 40% of ultimate.

A compilation of performance characteristics for Imbert and Biomass Corp. gasifiers is given in Table IV-12. These data were extracted from several sources for several different units but represent the best available information.

Table IV-13 lists manufacturers of downdraft gasifiers and the size, blast composition, number of operating units and status of development for the units that are available.

3. Stirred Bed Gasifiers

There are several gasifier designs that cannot be adequately described as being updraft or downdraft configurations because they rely to some extent on mechanical movement of the bed. These types have been classified as stirred bed gasifiers.

All of the stirred bed designs are pilot or demonstration stage of development. They have sufficiently different configurations that general statements of stirred bed gasifiers as a class cannot be made. Consequently, the following discussion treats each design separately.

a) EZ Manufacturing Co.

This company manufacturers a gasifier called the Pendu Gas System for conversion of dry sawdust and chips to low heating-value gas, char and wood tars. A single size is made having 3 GJ/h

Table IV-11

DATA SUMMARY FOR AN IMBERT GASIFIER WITH 65 kVA GENERATOR SYSTEM

Power (kW)	45	30	15	5
Wood Moisture Content, %	12.5	12	9	9.5
Fuel Loaded kg/h (dry)	52.0	43.7	29.1	22.1
Gas Production*, m^3/h	149.2	122.2	98.5	66.7
Fuel Gas Yield (m^3/kg dry fuel)	2.9	2.8	3.4	3.0
Average Gas Heating Value, MJ/m	5.44	5.16	4.59	5.26
Air* to Gasifier, m^3/h	79.2	64.0	60.5	36.5
Air* to Engine, m^3/h	137.4	111.2	104.3	63.2
Fuel Consumption, kg/kWh	1.2	1.5	1.9	4.4
Thermal Efficiency	78	72	74	79
Engine Efficiency	21	17	9	5
Overall Efficiency	16	12	7	4

* STP 15°C and 101.325 kPa (International Standard ISO 504)

Source: After Van Der Heijden et al.

Table IV-12

PERFORMANCE CHARACTERISTICS OF DOWNDRAFT GASIFIERS

CRITERIA	IMBERT	DUVANT	BIOMASS CORP.
State of Development	Commercial	Pilot	Commercial
Fuel: Size (cm)	1 - 10	5X10X20	1.3 - 5
Moisture Content (%, wb)	<20	<25	<30
Blast Composition	Air	Air	Air
Output Capacity[a] (GJ/h)	1 - 8	6 - 9	up to 15
Turndown	3 - 4 : 1	2 - 3 : 1	3 : 1
Cold Gas Efficiency[a] (%)	70 - 80	83 - 85	74
Gasifier Exit Temperature (°C)	375	600	350
Fuel Gas:			
Non Condensibles			
Yield (Vol. %, dry)			
N_2	52	51.4	42.7
CO	18.6	10.2	19.7
CO_2	9.4	16.4	12.7
H_2	19.1	20.4	17.5
CH_4	0.9	1.6	3.5
C_2+	(<.1)	N/A	2.1
HHV (MJ/m^3)	4.9		7.1
Condensibles Yield			
H_2O (kg/kg wet feed)	N/A	N/A	N/A
Tar (kg/kg dry feed)	very low	very low	very low
Particulate[d] g/m^3 (101 kPa, 15°C)	N/A	N/A	N/A
Power Required For Feeding and Gasification (kWh/GJ)	1.1[b]	5.8	---
Controls	Automatic	Automatic	Automatic
Ease of Operation	Fair	Fair	Fair
Response to Load Change	Good	Fair	Good
Equipment Cost ($/GJ/h)	2 GJ/h-$52,000[c]	9.3 GJ/h-$33,000[e]	3 GJ/h-$52,000 (U.S.)[d] 9 GJ/h-$30,000 (U.S.) 13 GJ/h-$25,000 (U.S.)

a - Based on higher heating value of gas at 101 kPa and 15°C.

b - No air blower.

c - Applies 60% of complete wood-to-electricity plant using diesel engine to gasifier and gas cleanup f.o.b. Bramerhaven, West Germany.

d - f.o.b. Yuba City, CA.

e - f.o.b. French Port. Add 5% for delivery to Eastern Canada.

Table IV-13

MANUFACTURERS OF DOWNDRAFT GASIFIERS

MANUFACTURER	BLAST COMPOSITION	OUTPUT CAPACITY (GJ/h)	OPERATING UNITS	STATUS OF DEVELOPMENT
Biomass Corp.	Air	up to 15	several	Commercial
Duvant	Air	6 - 9	1	Pilot (Model GA20)
Imbert	Air	1 - 8	several	Commercial
Pillard Inc.	Air	up to 7	several	Commercial
Pyrenco Inc.	Air	2 - 4	1	Pilot (uses wood pellets)

output, continuous feed automatic control and a dryer to reduce the moisture content of the feed.

Contrary to other designs, the wood is moved upwards through the reactor by action of an internal auger. Air is introduced at a level partway up the bed, while the produced char, oil and fuel gas are discharged above the bed. Feeding of the wood is done at the bottom with two diametrically opposed reciprocating piston feeders.

The dirty gas is cleaned in a wet scrubber. The recovered oil and char are dewatered and densified to produce a solid product.

The efficiency of converting the feed to cold fuel gas is claimed to be 75%.

b) Standard Solid Fuels

The gasifier being marketed commercially by Standard Solid Fuels uses one screw feeder to move feed material through a pyrolysis section followed by a second screw to move the resulting char through a gasification section (Figure IV-12). The evolved gases pass through a char bed that is kept hot by burning some of the char with air injected through mechanical bed aggitators. The char in this second chamber is replenished from the gasification section.

Composition data for the fuel gas are not available, but the heating value is claimed to be between 5.6 and 7.5 MJ/m^3.

The first installation of this gasifier design is at Taylor Clay Products, Salsbury, North Carolina. This system includes the gasification reactor, the char bed and gas cleanup equipment and is setup to feed fuel gas to brick kilns. Dry sawdust is the common feed material for this gasifier, although the manufacturer claims residue up to the size of hogged residues could be used.

A summary sheet for the standard solid fuels gasifier is given in Table IV-14.

The company is in the early stages of developing a system consisting of a dual fuel diesel engine connected to the gasifier. In the proposed system the fuel gas is cleaned in cyclones then is passed to a wet scrubber. After the wet scrubber, the gas will be compressed before piping to the engine.

Because of the char reactor used with the Standard Solid Fuels gasifier, the yield of tars from the system is expected to be comparable to that of a downdraft gasifier. The yield of water will depend substantially on whether the steam in the gas will react

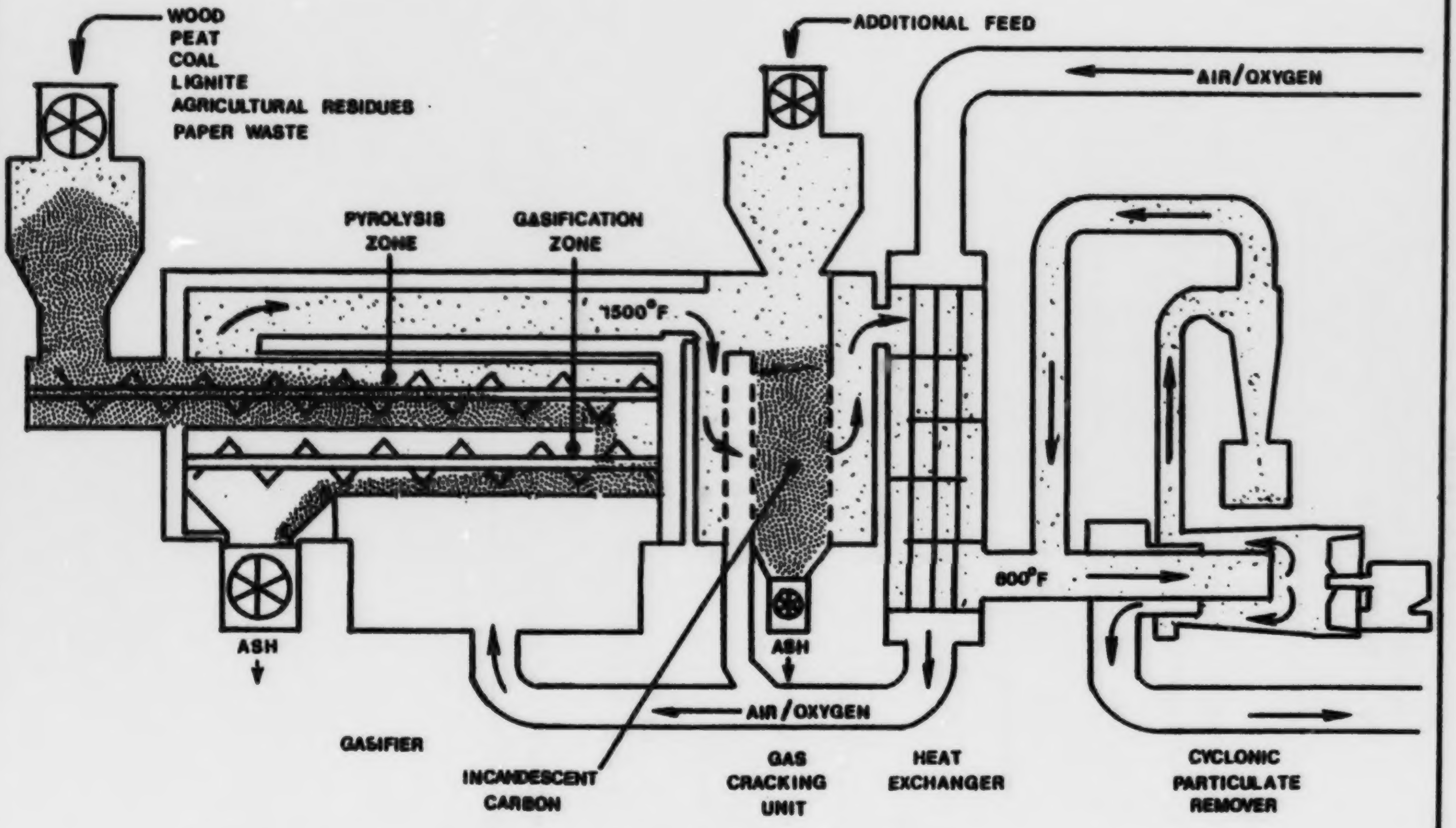


FIGURE IV-12 STANDARD SOLID FUELS GASIFIER

Table IV-14

PERFORMANCE CHARACTERISTICS OF THE STANDARD SOLID FUELS GASIFIER

CRITERIA	
State of Development	Demonstration
Fuel: Size (cm)	<1
Moisture Content (%, wb)	(dry sawdust)
Blast Composition	Air
Output Capacity (GJ/h)	20 - 70
Turndown	N/A
Cold Gas Efficiency (%)	70
Gasifier Exit Temperature (°C)	200
Fuel Gas: HHV (MJ/m^3)	5.6 - 7.5
Power Required For Feeding and Gasification (kWh/GJ)	2.5 - 3.5
Equipment Cost ($/GJ/h)	40 GJ/h-$15,000 (U.S.)

with the char in the char bed. If this happens, the heating value of the gas will increase, while the yield of condensible water will decrease. Insufficient data are presently available to assess the operating performance of the gasifier and char bed or to estimate the yield of condensible water and tars.

c) Wright-Malta Corporation

The Wright-Malta Corporation has developed a catalysed steam gasification process that exposes biomass to 2070 kPa steam at temperatures gradually increased to around 650°C. The biomass moves cocurrently with the steam and evolved gases through an externally heated kiln.

Studies have been conducted with two small gasifiers, one of batch-feed design, the other of continuous-feed design that transported residues with an auger. These experiments indicated that the best biomass is fast-growing open wood like poplar or non-woody biomass, with a moisture content of about 50%, wet basis. Alkaline carbonates were found to be the best catalysts, resulting in reduction of the discharged char to below 10% of wood input. Effectiveness of the catalyst increased linearly with concentration up to and above 10% by weight.

Near 90% of the fuel value and 95% of the mass feed can be converted to gas under optimum conditions, about 2070 kPa and 650°C. The pyrolytic tars and oils are completely cracked to fixed gases so that the gas evolved from processing wood at 50% moisture content consists of CH_4 - 29.5%, C_2H_6 - 5.8%, CO - 8.5%, CO_2 - 39.8% and H_2 - 16.4% on a dry volume basis. This gas has a higher heating value of 18 MJ/m^3. Operation at zero gauge pressure reduces the methane content and results in a gas of about 13 MJm^3 heating value.

Heat for the kiln is supplied by steam and heat recovered from the produced fuel gas.

A pilot reactor was built and operated in 1981. This unit used an internal auger to move solids through a 0.6 m diameter, 12 m long reactor. The best feed rate that could be achieved was 2.5 t/d, not the 6 t/d that was originally planned due to solids flow problems. Because of problems with solids flow through the reactor, the commercial unit that is planned will be a rotary drum, tilted to encourage flow of material.

The company plans to market commercial gasifiers having a feed rate of about 3.8 OD t/h using whole tree chips of 45 to 50% moisture content, wet basis, and minus 8 to 10 cm in size. The output of such a system is claimed to be near 75 GJ/h. The cost of this system is projected to be U.S.$4,500,000, or $60,000/GJ/h, including feeder, kiln, boiler and product gas cooler.

Because of the high heating value and low tar content of the produced gas, this system looks attractive for production of fuel gas for consumption in conventional prime movers, such as diesel engines and turbines.

4. Fluid-Bed Gasifiers

Fluid-bed gasifiers utilize a bed of either inert granular solids or ash and char particles that are fluidized by air or a mixture of steam and oxygen injected at the base of the bed through a distributor. The injected wood usually forms only a small fraction of the bed on a weight basis (<5%). Because of intense bubbling and mixing of the bed, the injected fuel is quickly distributed throughout the bed and bed temperatures are uniform. Fuel particles dry, pyrolyze and gasify as they move randomly within the bed, a vastly different situation from what occurs as particles move through the fairly spatially stationary reaction zones present in fixed-bed gasifiers.

The yield of char, condensible tars and fuel gas can be varied by adjusting the bed temperature, high temperatures favouring gas production, low temperatures favouring liquid and solid production. Typically, bed temperatures of 820° to 920°C are used for maximizing production of fuel gas, while bed temperatures of near 450°C are used for maximizing production of char. Intermediate temperatures lead to a maximum production of oils.

Figure IV-13 presents the energy content of the three product streams, gas, oil and char, produced by a fluid-bed unit manufactured by Energy Resources Co. Inc. of Cambridge, MA. These data represent what yields are expected from the 9 t/h infeed unit installed in Belle, Missouri to supply charcoal to the Kingsford Co. The feed consists of a bark, sawdust, wood chips and post peel mixture pulverized to minus 6 mm and dried from 50% to 25% moisture content, wet basis. Yield data for a wide variety of other biomass data have been obtained with a 1 t/h pilot unit at the company's headquarters that show gas yields as a fraction of feed generally increase with bed temperatures beyond 760°C, although data in Figure IV-13 show the energy content of the gases go through a maximum around 650°C.

Fluid-bed gasifiers need bed depths that are sufficient to maximize the production of fuel gas. Shallow beds do not allow sufficient time for cracking of tars.

Fluid-bed gasifiers can accept feed with up to a 55% moisture content, wet basis, although gas heating values decrease as feed moisture content increases. The commercial units that are available have operated primarily with feed moisture contents of 25% to 30%.

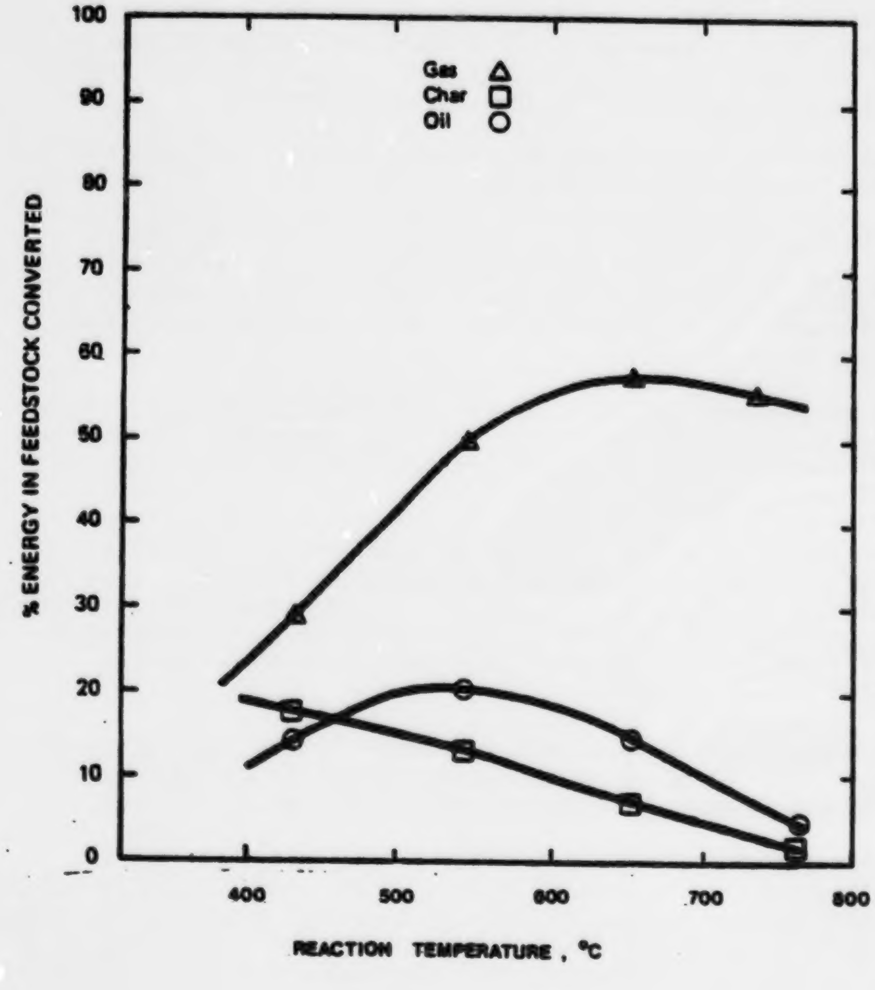


FIGURE IV-13 ENERGY CONTENT OF PRODUCTS FROM ERCO GASIFIER

Air-blown fluid-bed gasifiers operated for production of fuel gas generate a product containing some tar vapours, particulate from wood ash and the bed material, and having a higher heating value of between 4.8 and 5.6 MJ/m^3. Presence of tar vapours raises the gas heating value by less than 5%. Fluid-bed gasifiers contain less tar than updraft designs and probably more than downdraft designs.

Data obtained by Omnifuel Gasification Systems Ltd. show that the fuel gas produced from waste wood has a CO/H_2 ratio of 1.6 with a feed moisture content of 25%, wet basis, increasing to 1.9 when the moisture content is lowered to 11%. This indicates the minor involvement of evolved steam in the chemical reactions occurring in the gasifier. The observed CO/H_2 ratio is intermediate between the values found for updraft and downdraft gasifiers, although biased to some extent towards the updraft gasifier results.

The grate energy release rate for fluid bed gasifiers is in the range of 8 to 24 GJ/hm^2, well above the range determined for both updraft and downdraft gasifiers. The release rate is based on gasifier output as cold clean gas.

The air-to-fuel ratio of fluid-bed gasifiers can be more easily controlled than for any of the fixed-bed designs since fuel flow in the bed is not a hindrence. Data presented by Omnifuel Gasification Systems Ltd. and ERCO indicates that an air/fuel ratio of 2 to 3 is used for fluid-bed gasifiers. These are higher than those for fixed-bed designs.

Fluid-bed gasifiers available commercially have conversion efficiences of about 75% based on the heating value of the fuel gas, or upwards of 85% if the sensible heat of the fuel gas is included.

An example of a fluid-bed gasifier is the unit installed at Levesque Plywood Ltd. in Hearst, Ontario. This unit (Figure IV-14) was manufactured by Omnifuel Gasification Systems Ltd. and has a gas production capacity of 63 GJ/h with wood chips, bark, and sawdust fed at a moisture content near 30%, wet basis. Continuous operation of the gasifier began in May 1981, although the produced gas had to be flared for several months until four heaters and boilers were converted to burn the gas.

The gasifier is fed using a proprietary feeder produced by Stake Technology Ltd. that discharges into the bed against a moderate back pressure of about 35 kPa (gauge). Fuel gas is discharged from the gasifier at 17 kPa (gauge) and cleaned of particulate with two refractory-lined cyclones. Some of the sensible heat of this gas is used to preheat the incoming air but the temperature is purposely kept sufficiently high to prevent condensation of tars during piping to the dual fuel burners.

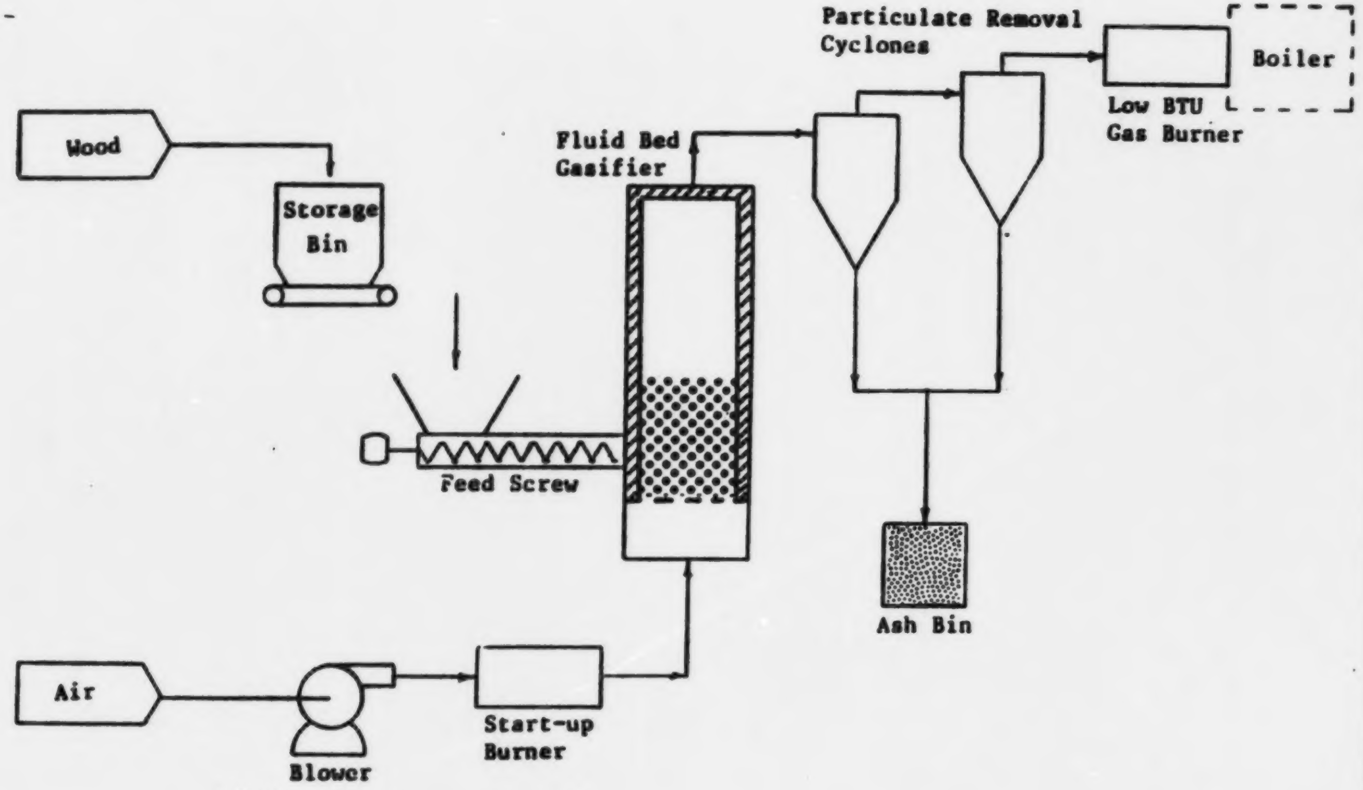


FIGURE IV-14 OMNIFUEL GASIFIER

The gasifier is fully instrumented for automatic operation and will shutdown automatically during abnormal conditions.

Several mechanical problems were encountered with the Hearst gasifier, related primarily to materials handling. These were: a bearing failure on the wood feeding system; wear of ash disposal system; and performance swings of the gasifier due to variation in feed moisture content. These problems have reportedly been resolved. The distributor design does not allow for removal of foreign material from the bed. This has made it essential that the fuel is clean.

A summary of performance data for fluid-bed gasifiers manufactured by ERCO and Omnifuel are presented in Table IV-15.

A list of the manufacturers of fluid-bed gasifiers, and the size, blast composition, number of operating units and status of development for the equipment sold by these manufactures is presented in Table IV-16.

E. Liquefaction Units

The term liquefaction as it is used here refers to all biological and thermochemical processes for conversion of solid wood to liquid fuels and chemicals. The possible conversion routes in schematic form are illustrated in Figure IV-15. Methanol is the only one of several liquid fuels that is produced by thermochemical conversion of wood to synthesis gas and subsequent chemical treatment. Higher alcohols are possible with fermentation processes.

The most immediate market for liquid fuels is in the transportation sector where it is essential to have a fuel that can be stored. Wood-derived liquid fuels are not attractive for electricity generation because of the numerous sequential process steps that are required for their manufacture and their low efficiency of conversion to electricity from wood feed, based on incoming chemical energy, when compared to alternative conversion routes.

At present, none of the liquefaction processes (Figure IV-15) has been commercially proven for wood. Two of the processes, methanol synthesis by gasification and fermention after acid hydrolysis, are presently technically feasible as a result of processes demonstrated with wood and other feedstocks. The other two processes, direct liquefaction and fermention after enzymatic hydrolysis, are at the research and development stage and are being intensely studied by numerous groups in Canada and abroad.

Table IV-15

PERFORMANCE CHARACTERISTICS OF FLUID-BED GASIFIERS

CRITERIA	OMNIFUEL	ERCO[a]
State of Development	Demonstration/ Commercial	Demonstration/ Commercial
Fuel: Size (cm)	<5	<0.6
Moisture Content (%, wb)	<50, <30 Preferred	<50, <25 Preferred
Blast Composition	Air	Air
Output Capacity[b] (GJ/h)	10 - 150	up to 100
Turndown	2 : 1	2.5 : 1
Cold Gas Efficiency[a] (%)	75	70 - 80
Gasifier Exit Temperature (°C)	760	N/A
Fuel Gas:		
Non-Condensibles		
Yield (Vol. %, dry)		
N_2	50 (est.)	58.4
CO	15.1	4
CO_2	16	17
H_2	10.3	55
CH_4	6.6	11
C_2+	2.0	4.1
HHV (MJ/m^3)	6.9	8.0
Condensibles		
Yield		
H_2O (kg/kg wet feed)	0.47	0.3
Tar (kg/kg dry feed)	N/A	0.02
Tar HHV (MJ/kg)	N/A	23.3
Particulate g/m^3 (101 kPa,15°C)	N/A	N/A
Power Required For Feeding and Gasification (kWh/GJ)	4.1	4.7
Control	Automatic	Automatic
Ease of Operation	Good	Good
Response to Load Change	Good	Good
Equipment Cost ($/GJ/h)	63 GJ/h - 13,000	

a - Operating in pyrolysis mode.
b - Based on higher heating value of gas at 101 kPa and 15°C.

Table IV-16

MANUFACTURERS OF FLUID-BED GASIFIERS

MANUFACTURER	BLAST COMPOSITION	OUTPUT CAPACITY (GJ/h)	OPERATING UNITS	STATUS OF DEVELOPMENT
Alberta Industrial Developments Ltd.	Air	50	1	Demonstration
Energy Products of Idaho Inc.	Air	2 - 250	None	Demonstration
Energy Resources Co. Inc.	Air	up to 100	1	Demonstration/ Commercial
Omnifuel Gasification Systems Ltd.	Air	10 - 150	1	Demonstration/ Commercial
Surlite	Air	12 - 85	1	Demonstration/ Commercial

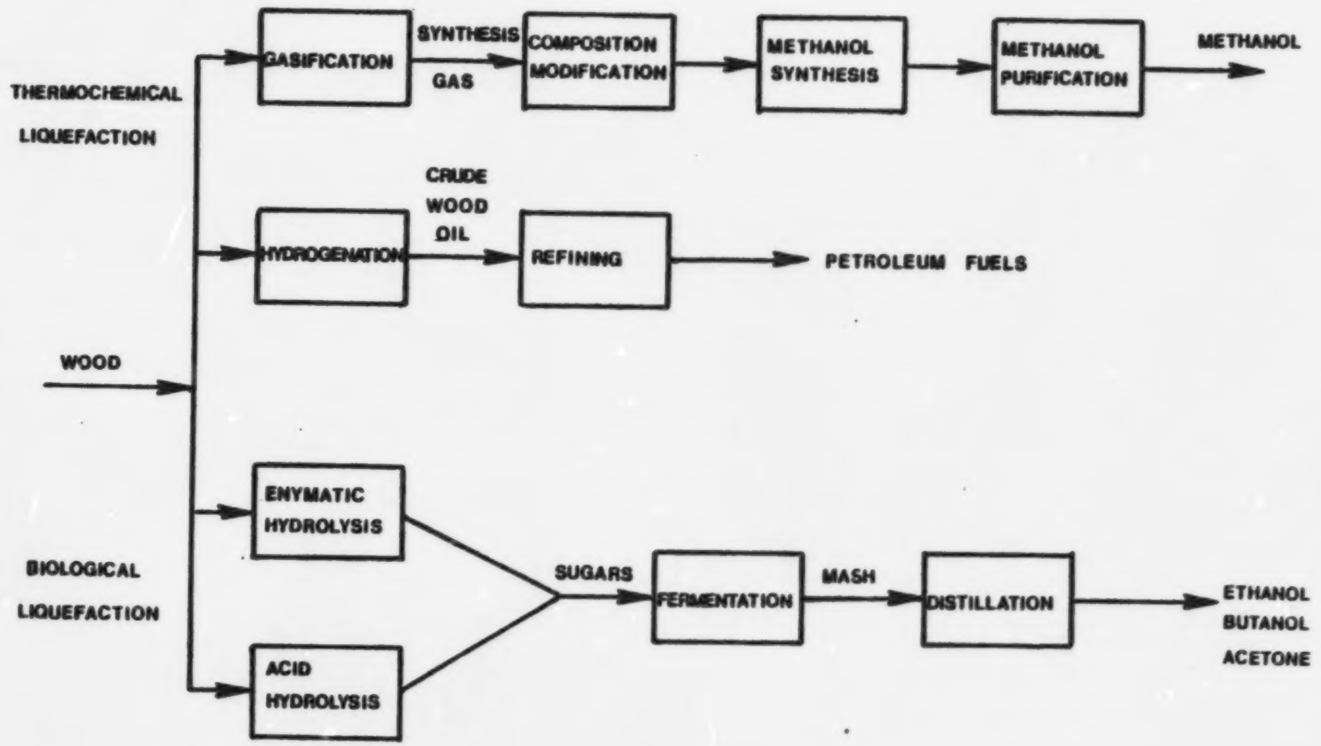


FIGURE IV-15 LIQUID FUELS AND CHEMICALS FROM WOOD BY BIOLOGICAL AND THERMOCHEMICAL PROCESSING

F. Pyrolyzers

By definition, pyrolysis is the thermal decomposition of a substance such as wood into char, liquids and gas by heating in the absence of oxygen. Commercial pyrolyzers usually use one of three possible methods of heating incoming wood: 1) heating indirectly through the wall of a reactor; 2) heating by direct contact with combustion gases from a separate combustor; and 3) heating by direct contact with combustion gases and hot solids as in a fluid-bed using some of the incoming feed as fuel. Only the first two of these configurations are true pyrolysis processes, although this distinction is not of practical importance since all designs attempt to maximize the production of char (30.2 MJ/kg). The byproducts of char production are a low heating value gas with a higher heating value of 6 to 8.2 MJ/m^3 and a heavy oil with a higher heating value of about 23.3 MJ/kg.

Pyrolyzers for wood residues operate at a temperature of about 450°C to obtain a char yield of 25 to 30% of the incoming feed, by dry weight. At this temperature about 85% of the higher heating value of the fuel is recovered in the products in the proportions: char, 60%; oil, 22%; gas, 18%. The evolved oil and gas are suitable as fuels. The oil is roughly equivalent to a No. 6 fuel oil in viscosity and chemical composition, although it has only about 80% of the heating value and may contain fine suspended char and inerts as produced. The fuel gas is similar to gas produced by gasification, but has a somewhat greater heating value and contains a higher proportion of hydrocarbons.

The fuel gas produced by pyrolysis could readily be used after clean-up for fuel to generate electricity in any system judged suitable for gas produced by gasification. The system would require enlarged clean-up equipment to cope with the greater amount of tars that would be produced. Because of poor alternate markets, the tar would likely have to be used for heating purposes or be returned to the pyrolyzer for cracking to fixed gases.

Pyrolysis systems emphasize production of char and, consequently, are not considered practical if electricity is to be a main product. Electricity could be generated using the gas as fuel, and the reader is referred to system configurations for gasifiers.

G. Steam Boilers

Historically, steam boilers have been the most important wood conversion units for generation of electricity from wood residues, and are expected to continue to be so for pulp and paper mills and integrated sawmills. They are available with several furnace configurations and with either water-tube or fire-tube boiler construction.

The numerous boiler designs that are presently available have evolved naturally in answer to a demand for improved operating characteristics and for flexibility in the range of steam pressures and temperatures. Because of the differences in boiler design, inclusive of feeding method, ash removal method and boiler tube layout, the available boilers have capital costs and inherent characteristics that make them best suited to certain steam capacities and steam pressures. Some designs are available as shop assembled and complete packaged units, others as shop assembled modular packaged units, while the largest sizes are only suitable for field erection. Table IV-17 classifies the principal boiler designs according to boiler configuration and furnace design, and indicates common ranges of capacity and typical steam conditions for each type. The capacities and steam conditions are based on industrial data and thus reflect both demand factors and cost of construction.

Package boilers are available in sizes up to about 30,000 kg/h steam. Larger units up to about 50,000 kg/h are available with modular construction to reduce the cost of field construction. These boilers were included as field erected boilers.

The range of capacities of the various boiler designs enables a preliminary match of a boiler to a particular application. For example, the travelling grate, semi-suspension burning design in the most common choice for boilers with a capacity in excess of 160,000 kg/h, and is often the preferred choice for capacities of 70,000 kg/h or more. Combustion cell designs which utilize a separate combustor connected to a boiler are the dominant choice for the lower end of the capacity range and for low to moderate steam pressures. These boilers are suitable for complete shop assembly.

Fluid-bed boilers with separate combustors, such as the units manufactured by Energy Products of Idaho, and with integral combustors, such as the units manufactured by Johnston Boiler Co., are an alternate choice for a package boiler.

Most package boilers are available as standard units for generating steam at low (400 kPa) and moderate (1,000 to 2,000 kPa) pressures. High pressure package boilers are available, but are often custom orders. Because of their design, most fire-tube boilers are available with steam pressures less than about 2,800 kPa.

Relative performance characteristics of most of the boiler designs are summarized in Table IV-18. These relative ratings support the preference for fire-tube designs for low to moderate steam pressures, low capacity, variable output and dry, sized fuel. Fluid-bed designs are relatively new to the boiler market and are presently best for sizes less than 50,000 kg/h steam capacity, low to moderate steam pressures, fairly steady output and wet fuel.

Comprehensive discussions of the various types of boilers and their performance when fired with wood residues are available in several references. Some of these references are Junge, Schwieger, Babcock and

Table IV-17

TYPES AND CAPACITIES OF STEAM BOILERS

BOILER TYPE	FURNACE TYPE	GRATE HEAT RELEASE LIMIT (GJ/hm^2)	CAPACITY RANGE		TYPICAL STEAM CONDITIONS	
			PACKAGED (kg/h)	FIELD ERECTED (kg/h)	PRESSURE (MPa)	TEMPERATURE (°C)
Water-Tube	Pile Burning: - Dutch Oven	8.5	---	20,000-90,000	6.2 4.1	480 400
	- Combustion Cells		5,000-34,000	---	1.7-2 4.1	Saturated or superheated 400
	- Inclined Grate	10.5	10,000-20,000	30,000-160,000	6.2 4.1	480 400
	Semi-Suspension Burning: - Pinhole Grate	7-11	10,000-30,000	30,000-100,000	6.2 4.1	480 400
	- Travelling Grate	11-13.5		50,000-320,000	6.2 4.1	480 400
	Fluid-Bed Burning	4.2	5,000-45,000	---	1-2 4.1	saturated or superheated 400
		FURNACE HEAT RELEASE LIMIT (GJ/hm^3)				
Fire-Tube	Suspension Burning	0.75-1.1	1,000-10,000		1-2.8	saturated or superheated

Table IV-18

RELATIVE* PERFORMANCE CHARACTERISTICS OF STEAM BOILERS

BOILER TYPE	HISTORY OF SERVICE	COMPATABILITY WITH WET OR UNSIZED FUEL	EASE AND COST OF OPERATION	EASE AND COST OF MAINTENANCE	STEAM PRESSURE AVAILABLE	RESPONSE TO LOAD CHANGE	TURNDOWN RATIO	CAPITAL COST
Water-Tube:								
- Pile Burning	G	G	G	F	G	P	P	P
- Semi-Suspension Burning	G	F - G	G	F	B	F	F	F
- Fluid-Bed Burning	G	B	G	F	F	G	P	G
Fire-Tube:								
- Suspension Burning	B	P	G	B	F - G	G	B	B

* Ratings are releative to other systems in the same major category: B = best; G = good; F = fair; P = poor.

Source: After Levi and O'Grady

Wilcox, B.H. Levelton & Associates Ltd. (1981), and Levi and O'Grady. Because of the general discussions available elsewhere, no discussion of the features of the boiler designs is repeated here.

The process of selecting a boiler for an electricity generating facility is an iterative and complex one that must consider details related to the boiler design and to the integration of the electricity generating facility with process steam and heat loads requiring high and low pressure steam. As a result, the boiler design must accommodate demands imposed by process steam requirements, including steam pressure, temperature and flow rate. The capacity of the boiler must be sufficient to accommodate process steam demand variations from typically 70% to 140% of average load, and electricity demand variations of up to 30% above average. If a back-pressure topping cycle is used for electricity generation, then high steam pressures are desired to improve cycle efficiency and allow the turbine discharge to be used as process steam. If there is very little demand for process steam and the main function of the boiler is for the generation of electricity, a condensing turbine may be desireable. High steam pressures improve the cycle efficiency and reduce steam requirements, reducing the condenser heat removal. From these considerations, it is apparent that the boiler must be sized as part of a cogeneration system to satisfy the steam requirements of the power turbine and the mill processes.

Costs of boilers are difficult to generalize because of the influence of site specific design factors. As outlined previously, there are two basic cases: 1) a boiler may be installed for electricity generation where there is only a small demand for steam heating; or 2) a field erected boiler may be installed for cogeneration of electricity and process steam. Both designs may have capacities up to and exceeding 5 MW, but there would be a great difference in process steam production.

Figure IV-16 presents typical capital cost factors for package boilers with steam pressures up to about 1,700 kPa. The cost factors account for:

- basic unloading equipment for incoming fuel
- fuel processing equipment
- limited storage capacity with unloading equipment
- boiler and auxiliaries
- pollution abatement equipment
- control system for all of the above

and have been updated from 1979, the date of compilation, to October 1981 using Marshall and Stevens Indices. These cost factors are in general agreement with data obtained from Canadian suppliers of package boilers. Costs increase above the indicated values if highly automated systems are required. Installed costs may run from 30 to 60% above the equipment cost of a package unit.

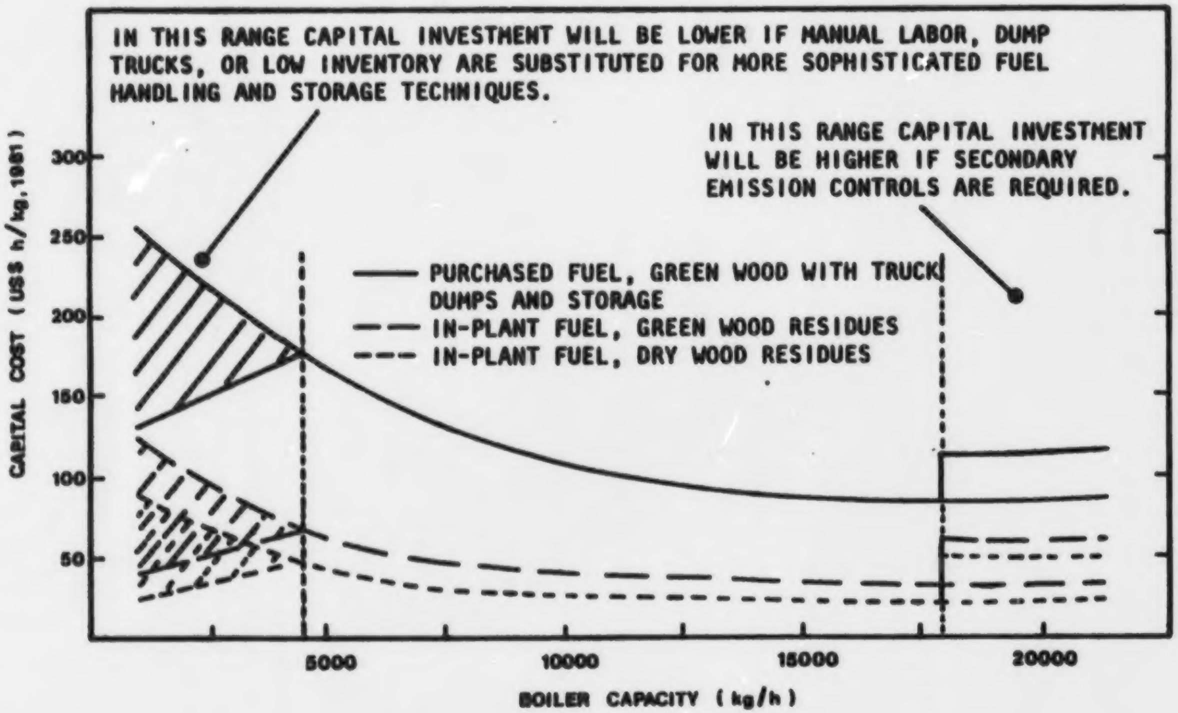


FIGURE IV-16 CAPITAL COST AS A FUNCTION OF BOILER SIZE

SOURCE: AFTER LEVI AND O'GRADY

In a study of coal boilers, of both packaged and field erected design, Coffin established that addition of superheat adds 1.45% to the cost per 1,000 kPa pressure above a base pressure of 1,720 kPa (gauge). Hence, a boiler for generating steam of 4,130 kPa would cost 3.5% more than a boiler for generating steam of 1,720 kPa. Coffin also observed that the cost of two units sized to 50% of the desired capacity cost 10% more than a single unit of equal size, and that a field erected unit cost nearly twice the cost of a package unit.

General costs of modular and totally field erected units are difficult to estimate except in very approximate terms. A detailed cost estimate has been conducted by Archibald and Gabriel for a wood-fired boiler and their results, shown in Table IV-19, were updated to October 1981. These data indicate boiler costs increase by 4.8% per 1,000 kPa above 1,720 kPa (gauge), considerably higher than the 1.45% estimated by Coffin. Boiler plant costs increase by 6% per 1,000 kPa above 1,720 kPa (gauge). The boiler used in the example is suitable for generating either 2.5 MW using a back-pressure turbine operated with a discharge at 1,720 kPa (gauge), or 5 MW using a condensing/extraction turbine, with extraction at 1,720 kPa (gauge).

The costs estimated by Archibald and Gabriel are somewhat below the cost of a 54,000 kg/h, 5,860 kPa, 460°C boiler installed for cogeneration of 5 MW of electricity and process steam at a large forest products mill in Wright City, OK (Vraspir and Crane). The cost in 1979 for the boiler plant complete with tube and tile, pinhole grate boiler, air heater, clarifier, gravity filters, two demineralizer trains and condensate polisher, and all pumps and piping was $7.2 million. The equivalent cost in October 1981, would be $9.93 million, or $180/kg/h, about 35% higher than the estimates by Archibald and Gabriel for a lower pressure installation, although the two units probably have numerous different site-specific design features.

Vraspir and Crane state that boiler costs increased dramatically for units with steam pressures above 5,860 kPa (gauge). Matschukat and Fischer indicate that a substantially more sophisticated water treatment system is necessary for boilers operating above 4,130 kPa (gauge), leading to higher plant costs.

Comprehensive lists of the suppliers of packaged and field erected boilers are given in the annual and directory of Pulp and Paper Canada and several other trade directories.

Table IV-19

COST OF A FIELD ERECTED BOILER PLANT

	CASE NO. 1		CASE NO. 2	
BOILER CAPACITY	68,000 kg/h		68,000 kg/h	
STEAM CONDITIONS	1,720 kPa (gauge) 230°C		4,130 kPa (gauge) 316°C	
	Oct/81 $ X 1,000*	%	Oct/81 $ X 1,000*	%
Fuel Handling	2,290	($34/kg/h)	2,290	($34/kg/h)
Boiler Plant				
- Site, Foundation, Building	890	12.8	890	11.1
- Boiler	3,300	47.4	3,680	46.2
- Piping, Insulation	700	10.0	760	9.6
- Instrumentation	420	6.0	480	6.0
- Electrical	280	4.1	320	4.0
- Water Treatment	110	1.5	500	6.4
- Scrubber	825	11.8	825	10.4
- Auxiliaries (pumps, deaerator)	445	6.4	510	6.4
		100		100
	6,970	($103/kg/h)	7,965	($117/kg/h)
TOTAL:	9,260		10,255	

* Marshall and Stevens Index Ratio for 1981/1979 is 1.27.

Source: After Archibald and Gabriel.

V POWER FLUID HEAT EXCHANGE UNITS

Equipment that transfers heat from a primary working fluid produced from a conversion unit to a secondary working fluid for use in a prime mover is essential to several of the topping cycles for generation of thermal and electrical energy from wood. Bottoming cycles use a boiler to transfer heat from the discharge of a prime mover or some similar high-grade waste-heat source and produce steam or organic vapour for either electricity generation or thermal uses. In either case, these heat transfer devices have been classified as power fluid exchange units.

A. Heat Recovery Boilers

Heat recovery boilers may be applied to generate steam or organic vapours, although the former working fluid is customary. These boilers may be applied with either combustion devices for production of low and high pressure steam for electricity generation or with waste heat streams, usually for production of process steam. Organic fluids have applications for electricity generation using low-grade heat recovered from waste streams. Organic fluid systems for power generation use a Rankine cycle configuration with a vapour turbine of similar design to a steam turbine.

1. Steam Boilers

Heat recovery boilers coupled directly to a primary wood conversion unit, such as a combustor, were considered previously in the discussion of package boilers that use a combustion-cell furnace. Typical layouts of these boiler systems are illustrated in Figure V-1. The reader is referred to the previous section on steam boilers within the chapter on wood conversion units for the characteristics and costs of package boilers.

Heat recovery boilers may be of fire-tube or water-tube design. Fire-tube units are limited to steam capacities below about 10,000 kg/h and are available in standard units for low and moderate pressures, although they can be used to generate steam at pressures up to about 6,900 kPa (gauge). High pressure steam at higher flow rates requires use of water-tube boilers. The upper limit of pressure for these boilers is near 10,300 kPa (gauge).

Inlet gas temperatures for heat recovery boilers containing ash particles are limited to 1,000 to 1,100°C to minimize problems with ash slagging and fouling of boiler tubes. Since the boiler is mounted externally, heat transfer is by convection only. In integral boilers, radiative heat transfer makes a substantial contribution to total heat transfer and, consequently, the boiler section can be smaller in area than if only convective transfer occurred, as in a heat recovery boiler. To compensate for loss of

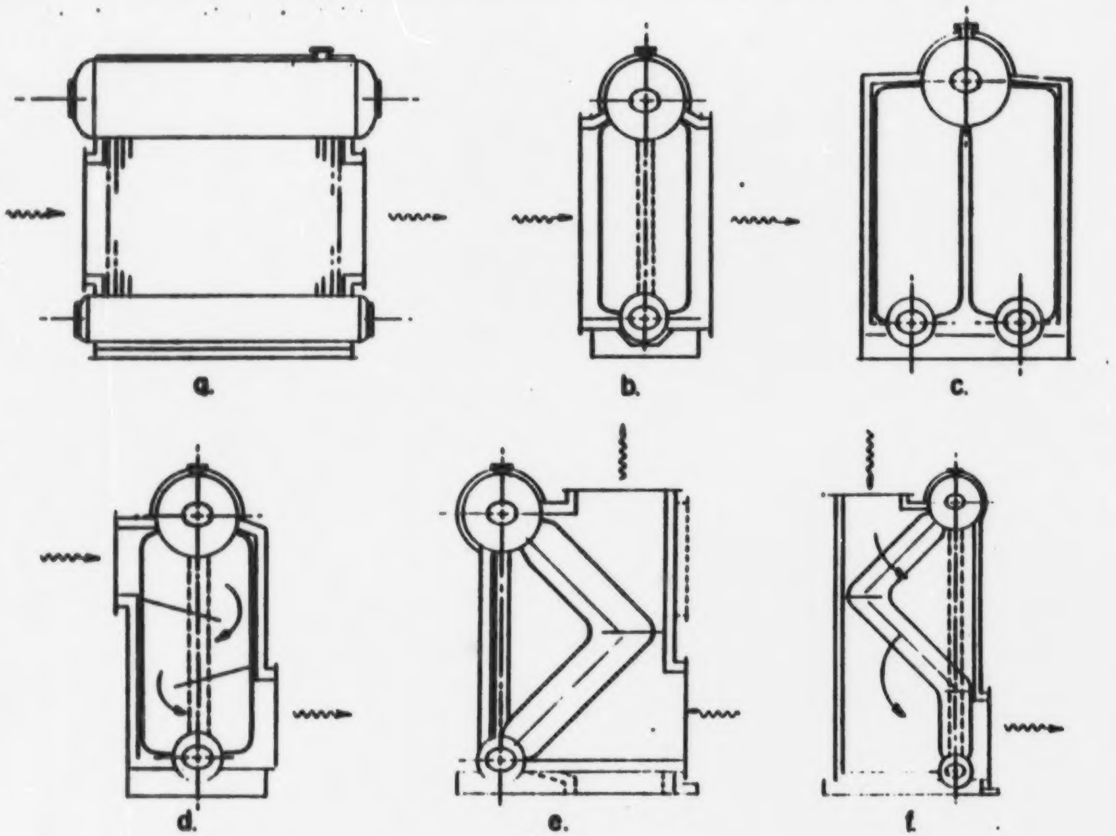


FIGURE V-1 LAYOUTS OF WATER-TUBE HEAT RECOVERY BOILERS

radiative heat transfer, heat recovery boilers often utilize finned tubes in the superheater, boiler and economizer sections, but not in the air-heater section.

Specification of the particulate loading and gas velocity exiting a combustor is essential for proper design of a heat recovery boiler. A design guide for gas velocities in heat recovery boilers is given in Table V-1. Gas velocities should not be too low (<11 m/s) in water tube designs, since this promotes deposition on the tubes.

Heat recovery boiler/combustor combinations simplify field erection because of their modular construction. This is a definite advantage over field erection boilers up to about 50,000 kg/h steam production. Although no experimental data are available, the thermal efficiency of a heat recovery boiler/combustor system will be comparable to or slightly below the thermal efficiency of integral boilers because of possible higher surface heat losses. Thermal efficiencies between 65% and 75% can be expected, depending on the moisture content of the fuel, excess air and stack gas temperature. Efficiencies in the lower part of this range are typical for conventional power boilers firing wet fuel.

Utilization of heat recovery boilers to capture heat from the exhaust of turbines and diesel engines is proven technology. This system design could readily be applied to produce process steam from gases discharged from prime mover equipment used to generate electricity. The boilers are available with a full range of steam conditions.

2. Organic Fluid Boilers

Suppliers of heat recovery boilers designed for industrial organic fluids could not be located. The work which has been conducted on organic Rankine cycles has been primarily at a research and development level to determine the performance characteristics of numerous working fluids, the more common of these being halocarbons (refrigerants), hydrocarbons and ammonia. The motivation for development of organic Rankine cycles has been for high efficiency, compact electricity generating systems needed by the military and for space travel, and for high efficiency utility power generating plants. The attractiveness of these cycles is their potentially high conversion efficiencies (+20%) at relatively low operating temperatures (340 to 370°C) (Leighton). The capacity of these power cycles has so far ranged from 0.1 to 1,000 kW.

Organic Rankine cycles can be used to recover heat from hot gas streams in the range of 90°C to 400°C with presently identified fluids. Most of the fluids have optimal cycle performance between 150°C and 230°C. One of the more promising of the organic fluids

Table V-1

HEAT RECOVERY BOILER DESIGN GUIDELINES

GAS CONDITIONS	MAXIMUM GAS VELOCITY (m/s) FIRETUBE	WATERTUBE IN-LINE	WATERTUBE STAGGERED	FOULING FACTOR	MAX. FIN/INCH	FIN THK (mm)	SOOT-BLOWERS
Clean Comb. products of natural gas	60	40	30	0.001	6	1.3	No
Average Comb. products of #2 oil & wood, fume incinerators	50	30	25	0.002	5	1.3	Provision
Dirty Comb. prod. of #6 oil, liquid incinerators, municple sludge, fluid cat cracker	40	25	20	0.004	4	1.5	Yes
Dirty & Abrasive Comb. products of coal	30	20	10	0.004	3	1.9	Yes

Source: CSATHY

for heat recovery from waste streams having a temperature near 300°C is ammonia. At a turbine inlet temperature of 316°C, the ammonia cycle has a higher efficiency than a steam cycle operated at 2,000 kPa and 316°C, but turbine inlet pressures must be in excess of 6,900 kPa. Operating at a pressure of 13,800 kPa results in a cycle efficiency for ammonia 33% greater than for steam. Such high pressures would require special boiler design appropriate to the thermodynamic properties of ammonia.

The only equipment that has been well proven for organic fluids is the equipment used in refrigeration cycles. These use freon, ammonia, carbon dioxide, air, paraffinic hydrocarbons and numerous other fluids. These cycles operate between 15°C and 38°C and utilize reciprocating and centrifugal compressors. The evaporators (boilers) used in these cycles are simply heat exchangers and are not on the scale of industrial steam boilers.

The main hindrence to development of boilers for organic fluids is the lack of thermodynamic data. In additon, special consideration must be made during design for any safety hazards that the fluid may cause for workers in the case of leakage. Hydrocarbons are flammable and could be a fire hazard, while many of the other candidate fluids are toxic and thus require very special handling considerations.

Substantial demonstration of organic Rankine cycle heat recovery is needed before an assessment of its viability in forest products industries is possible, especially concerning safety hazards.

B. Hot Gas/Hot Oil Heat Exchangers

Heat exchangers may be used for recovery of heat in hot gases by transfer of this heat to a hot oil heating system. Such an application would be for waste heat recovery as part of a cogeneration system rather than as a means of generating electricity. Heat exchangers used in bottoming cycles for vapourization of organic fluids are discussed in the preceding section covering heat recovery boilers.

Several companies market heat transfer oil suitable for operation with bulk temperatures up to 400°C, including Dow Chemical Co., Mobile Chemical Co. and Monsanto Co. These fluids are flammable, but not toxic, although reasonable care must be exercised during handling and disposal.

Hot gas/hot oil heat exchangers for use at oil temperatures possible with commercial oils are commercially available either as standard units or as custom designs. Carbon steel tubing is commonly adequate up to a temperature of about 370°C when under continuous load. Higher temperature operation requires use of steel alloys, ranging from chromium steels at intermediate temperatures to stainless steels at high temperatures (>650°C).

Hot oil systems run at low pressure and are used in the forest products industry for drying applications such as lumber kilns and veneer dryers. Combustion systems such as the type manufactured by Konus-Systems Inc. are available for direct heating of hot oil. These particular units are available in sizes from 4 to 25 GJ/h for firing with dry and green shavings, hog fuel and other residues.

For power generation systems the sole application of hot gas/hot oil heat exchangers is for recovery of heat from hot gases discharged from prime movers, such as turbines, internal combustion engines, etc. for heating and drying in other areas of the plant. These are very attractive applications for prime movers such as gas turbines and diesels that do not use steam as a working fluid, since manning of the system is not governed by provincial steam regulations and totally automated systems or ones requiring minimal supervision could be operated.

C. Hot Gas/Hot Air Heat Exchangers

1. Applications

A hot gas/hot air heat exchanger is an essential component of a power generating system utilizing an indirectly-fired gas turbine. In these systems a non-pressurized combustor is used to generate hot combustion gases which, with the use of a heat exchanger, heat a pressurized secondary working fluid, most commonly air, that is then expanded through a gas turbine to generate electricity. A compressor with the desired pressure ratio supplies the incoming air to the heat exchanger.

An important advantage of this system design is that clean air is the working fluid for the turbine, avoiding corrosion and erosion of internal components that would occur with either direct use of dirty gas produced from a pressurized combustor, or, alternately, use of a wood-derived fuel gas containing fine particulate and tar.

The hot gas/hot air heat exchanger may be placed in the exhaust gas duct or, when a fluid-bed combustor is used, in the bed itself. If the heat exchanger is placed in the exhaust gas duct, combustion gases are discharged from the heat exchanger at a much reduced temperature. This exit temperature at its lowest must be sufficiently above the dew point temperature to prevent any condensation in downstream ducting. A heat recovery boiler may be used downstream of the heat exchanger, if steam is needed.

If the heat exchanger is placed in the bed area of a fluid-bed combustor, then the high temperature air and high temperature combustion gas can be produced simultaneously. This design takes advantage of high heat transfer coefficients in the bed, about 280 W/m^2°C, that are four to six times those common for heat transfer surfaces exposed to gas flow only.

The two locations for the hot gas/hot air heat exchanger lead to correspondingly different system performance. For the fluid bed installation, any increase in heat removal from the bed for power generation results in increased heat output in the flue gas, since the bed temperature must be held nearly constant to achieve complete combustion. If the heat exchanger is installed in the flue gas and it is of sufficient capacity, an increase in heat removal for power generation will only modestly alter the heat flow in the exiting flue gases.

Data for operating conditions of gas turbines are given in Chapter VI. Industrial turbines commonly use inlet temperatures of 980°C and have compressor pressure ratios of from 4 to over 12. The optimum pressure ratio is a function of the inlet temperature of the turbine and, for the indirectly-fired case, the outlet temperature of the heat exchanger.

The upper limit for the temperature of the combustion gases entering the heat exchanger is fixed by the softening point of the particulate entrained in the combustion gases. Based on data obtained from combustor manufacturers, the preferred temperature of the combustion gases is 980 to 1,100°C for suspension and multiple chamber designs, while fluid-bed temperatures are commonly 870°C. The temperature drop from the gas side to the air side of a heat exchanger operated at these temperatures has been determined by Marksberry and Lindahl to be as high as 120°C. This means the maximum exit air temperature that could be expected in a counter-flow heat exchanger is between 730°C and 980°C, depending on the combustor selected.

Whether these high temperatures can be achieved depends on the temperature limits of the heat exchanger materials and heat loss from ducting carrying the compressed air to the turbine inlet.

No commercial indirectly-fired turbine systems are presently operated. Numerous engineering analyses of these designs have been done for coal-fired combustors.

The serious negative aspect of the system design is the operation of a heat exchanger at temperatures up to and beyond the limits for common steel alloys. In addition, all hot gas ducting has to be refractory lined to protect strutural steel parts and minimize heat loss.

2. Materials Selection

Maximum thermal efficiency of a gas turbine is achieved at maximum turbine inlet temperature. In an industrial-fired turbine, the inlet temperature is dependent on both the combustor operating temperature and the effectiveness of the heat exchanger. The heat exchanger design must balance the improved turbine efficiency achieved with increased gas and air temperatures against the higher cost and poorer durability of materials capable of operation at these elevated temperatures.

As small a temperature drop as practical is desired between the hot-gas side and the air-side of the heat exchanger. The log mean temperature difference across the heat exchanger depends on the overall heat transfer coefficient and, thus, gas velocities, as well as the surface area of the heat transfer surface. The design must balance several factors including limitations on external gas velocity to avoid erosion and deposition, cost of increased heat transfer surface, allowable operating temperature of heat transfer tubing and supports, and allowable pressure drop across the heat exchanger.

The most influential operating condition for the design of the heat exchanger is the temperature of the heat transfer tubing. Assuming an external gas temperature of 980°C to 1,100°C and an exit air temperature of 760 to 870°C, the tube metal must operate at a temperature of 860°C to 980°C. These temperatures are well beyond the 540°C to 570°C upper limit set for most superheater tubes used in boilers producing steam at pressures up to 27.6 MPa. To operate at these temperatures, superheater tubes are usually constructed of expensive austenitic stainless steels. As temperatures decrease, increasing use is made of chromium/molybdenum and low-carbon steels. Babcock and Wilcox state the maximum allowable temperature of Type 304 and Type 321 austenitic stainless steels is 760°C. The Metals Handbook presents data indicating the upper limit for use of austenitic steels falls near 870°C. Comparing these temperatures to those desired for optimum operation of the power cycle clearly shows that either lower turbine inlet temperatures must be used to avoid materials problems, or other more suitable alloys or ceramics must be incorporated into the high-temperature section of the heat exchanger. With use of very expensive nickel base alloys, it is possible to extend the temperature limit to upwards of 980°C, while use of ceramics increases this temperature to 1,500°C. Both of these classes of materials may not be practical or economical for use in present-day forest products mills.

Marksberry and Lindahl surveyed the metals available for in-bed heat exchangers for fluid-bed coal combustors and concluded that a realistic limit on tube temperatures is 700°C, this being a modest increase over performance materials in boilers.

Considering the heat exchanger metals that are available, it is apparent that the practical limit of tube temperature for a gas/air heat exchanger is 700°C to 760°C. Such temperatures could lead, with well insulated ducting, to turbine inlet temperatures in the vicinity of 600°C to 700°C.

Hague International of South Portland, Maine, markets shell and tube style ceramic heat exchangers for use as high-temperature heat recuperators suitable for gas temperatures to 1,500°C. Bonded silicon carbide is used to form the tubes which are mounted in special spring-loaded fittings to allow for thermal distortion. Fittings are available for internal pressures up to 1,000 kPa.

Considerable success has been achieved with low-pressure (<70 kPa gauge) units installed to recover heat from gases exiting steel forge furnaces. The heat exchangers use 1.2 m long finned and plain tubes in single- and double-pass configurations. The tubes can withstand cyclic temperatures and are tolerant to flyash. Material service lives are expected to exceed five years. Annual maintenance of refractory lining of the heat exchanger duct may be necessary.

3. Manufacturers

Many companies are capable of fabricating heat exchangers for industrial applications, although high temperature heat exchangers are not standard items. Manufacturers of heat exchanger equipment are listed in the Pulp and Paper Annual and Directory, Chemical Engineering Equipment Buyers Guide, and other trade directories.

a) Fluidyne Engineering Corp.

Fluidyne Engineering is a general engineering company involved in design of control systems, pollution and environmental engineering equipment and energy systems. They have a pilot-scale fluid-bed combustor that has been fired with coal for testing of tube alloy steels. Some combustion tests have also used wood waste fuel.

The company markets fluid-bed air heater systems suitable for process heat supply and indirect turbine power systems. The heat exchanger is a bayonet design cantilevered from the wall in the fluid bed.

The power requirement of the main air supply for the combustor is 1.5 to 2.0% of gross heat output.

These air heaters are available as skid-mounted units in sizes from 5 to 20 GJ/h. The maximum metal temperature in the heat exchanger is 760°C.

b) Hague International

Shell and tube shop-assembled ceramic heat exchangers (CERHX) are marketed by this company in standard sizes to 100 GJ/h. Over twenty-five units of 3 to 14 GJ/h size have been installed to date for heat recovery from gases with temperatures of 1,200 to 1,400°C (Bjerklie et al).

The tubes come with either plain or finned exteriors and have one of two compositions, depending on whether internal pressures are less than 70 kPa (gauge), or 70 to 1,000 kPa (gauge). Special spring-loaded connectors allow easy tube removal by mill personnel, yet can be made leak-free.

The ceramic used is silicon carbide. This material is available with one binder for continuous use at temperatures up to 1,300°C, while with another binder, it may be used to 1,500°C.

The temperature drop from the gas inlet (tube exterior) to the air exit (tube interior) depends on the inside and outside surface heat transfer coefficients. One example is a unit with a gas temperature of 1,288°C and a two-pass design. With an inlet air temperature of 20°C, the air exit temperature is 870°C, a temperature drop of 418°C. The two-pass designs have effectiveness values of 60 to 70% when operated on steel forge furnaces to preheat combustion air.

Assume conditions typical of an indirectly-fired gas turbine: the heat exchanger effectiveness is 70%, the shell-side inlet gas temperature is 1,000°C; and the inlet tube-side air temperature is 20°C. Under these conditions the turbine inlet temperature would be about 700°C, neglecting any heat losses.

No installations of the heat exchangers have been made for installations using internal pressures greater than 70 kPa (gauge). These high-pressure units are under development to prove-out sealing capability, to lengthen tubes to 2.4 m and to establish service life. Costs are determined for individual cases only. They intend to standardize the units in the same manner as done for the low-pressure units.

VI PRIME MOVERS FOR ELECTRICITY GENERATION.

A. Steam and Organic Vapour Turbines

1. Cycle Configurations

The most common topping cycle operating in the forest industry is the steam boiler-steam turbine generator system. Steam is generated in the boiler and is expanded through the turbine producing electricity at the generator. Low-pressure steam discharged from the exhaust of the turbine is then piped to process for thermal energy applications. To minimize heat losses in the system, the steam condensate is returned to the boiler after utilization by the process.

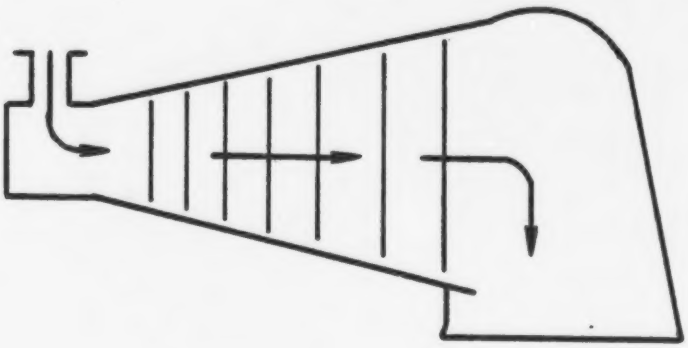
A Rankine bottoming cycle using steam or an organic vapour as the working fluid consists of an fluid boiler, a vapour turbine, a regenerator and a condenser. The liquid working fluid is vaporized at constant pressure in the boiler using waste heat from process or an existing topping system. The hot vapour is expanded through a turbine-generator set, passed through a regenerator and condenser and then pressurized by a feed pump, preheated and returned to the vapour generator.

Currently, commercially available Rankine bottoming cycles use steam as the working vapour. Organic Rankine cycles operate at lower temperatures than steam cycles, 100° to 300°C, compared to 200° to 550°C, and can recover low temperature waste heat more effectively than steam cycles (Noyes). This possible application has resulted in continuing research to raise the maximum operating temperatures and improve system efficiencies.

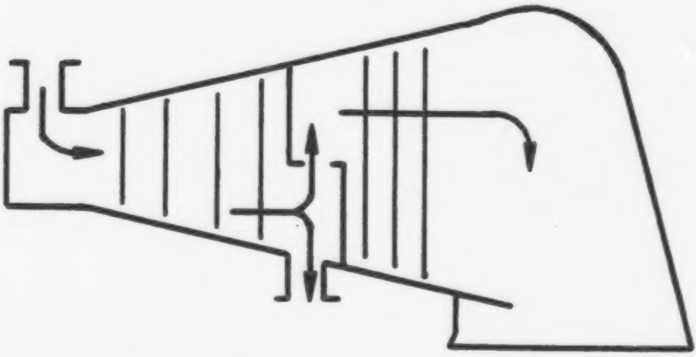
2. Performance Characteristics

Steam turbines can be separated into two basic categories: condensing turbines which exhaust steam at sub-atmospheric pressures; and back-pressure (non-condensing) turbines which exhaust steam at a higher than atmospheric pressure. While back-pressure turbines may be either single or multiple-stage, condensing turbines generally have more than one stage of blades.

In addition to the two basic categories, steam turbines may be further classified by their steam flow pattern. Straight-through turbines use full throttle steam flow from inlet to exhaust (Figure VI-1). Extraction turbines (Figure VI-2) have one or more locations where intermediate pressure steam may be withdrawn from the turbine allowing for multiple-pressure process steam. Induction turbines operate in a manner opposite to extraction turbines. Steam at lower than throttle pressure is injected into the turbine

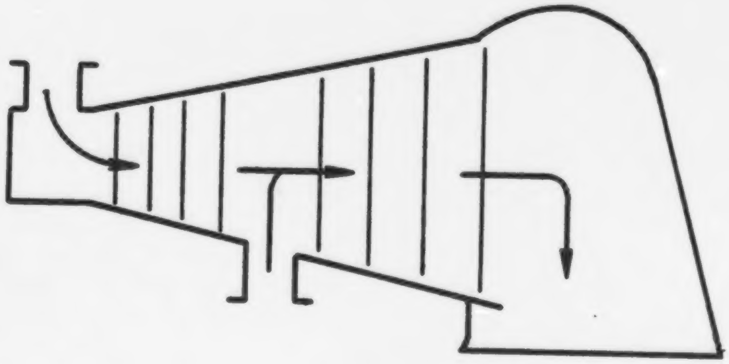


A. STRAIGHT-THROUGH FLOW

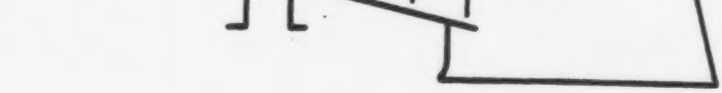


B. EXTRACTION FLOW

FIGURE VI-1 STRAIGHT-THROUGH AND EXTRACTION FLOW TURBINES



A. INDUCTION FLOW



B. COMBINATION FLOW

FIGURE VI-2 INDUCTION AND COMBINATION FLOW TURBINES

downstream of the inlet to generate additional power. Turbines are also available which combine both the extraction and induction functions (Figure VI-2). In this case, the intermediate pressure steam line is kept at constant pressure extracting or admitting steam as required.

The amount of power generated by a steam turbine is a function of the turbine shaft speed which is dependent on the steam flow rate. In the case of back-pressure turbines, the steam flow is controlled at the turbine inlet using a governor which opens and closes valves, admitting steam to the turbine. The valves may be arranged as spring loaded nozzle valves which open in sequence as turbine load increases, or as a single throttle valve regulating the whole steam load. In the case of extraction and induction turbines, further control of steam flow is achieved by regulating the amount of intermediate pressure steam admitted to or extracted from the turbine. The amount of control of steam flow through the intermediary port is generally tied to the steam demand of the process.

Another control device of importance found on steam turbines is the overspeed trip which prevents damage to the turbine caused by excessive speed. A typical overspeed trip may consist of a spring-loaded bolt fitted radially into the end of the shaft. At a certain speed, determined by the spring constant, centrifugal force will overcome the spring force and the bolt will move outwards, activating the overspeed trip lever which closes the inlet steam valve.

In addition to those devices above, other instrumentation associated with turbines include: low bearing-oil-pressure trip, shaft-vibration meter and steam condition (temperature and pressure) meters.

Overall turbine performance is measured by various parameters depending on turbine type, flow pattern and application. The theoretical steam rate or steam consumption in kilograms per kilowatt hour output under isentropic expansion is found by the relationship:

$$\mathrm{TSR} = \frac{3600}{h_1 - h_2}$$

where: 3600 = ideal isentropic thermal to electrical conversion (kJ/kWh);
h_1 = steam enthalpy at inlet condition (kJ/kg);
h_2 = outlet steam enthalpy after isentropic expansion (kJ/kg).

Note that the outlet steam enthalpy, h_2, is found at conditions of actual outlet pressure and inlet entropy. The theoretical steam rate is dependent only upon inlet pressure and temperature and outlet pressure and is the same for all turbines under the same conditions.

Actual steam rate (ASR) is the actual steam consumption rate per unit output (kg/kWh) and is found by:

$$ASR = \frac{3600}{h_1 - h_{2a}}$$

where: h_1 = steam enthalpy at inlet conditions (kJ/kg);
h_{2a} = steam enthalpy at actual outlet conditions (kJ/kg).

Overall turbine efficiency can be determined by dividing the theoretical steam rate by the actual steam rate, thus comparing the actual turbine performance to that of an ideal turbine of the same design and operating with the same inlet conditions.

The actual steam rates (and therefore turbine efficiencies) depend on steam conditions and turbine design, i.e. size, shaft speed, number of stages and leakage loss (Table VI-1). Typical actual steam rates and turbine efficiency ranges for various types of nonextraction turbines are shown in Table VI-2.

Another useful parameter to measure turbine performance is the heat rate (HR), the amount of thermal energy used by the turbine per unit energy output (kJ/kWh) neglecting gear and generator losses, which can be determined by:

$$HR = ASR\,(h_1 - h_{2a}')$$

where: ASR = actual steam rate (kg/kWh);
h_1 = steam enthalpy at inlet conditions (kJ/kg);
h_{2a}' = steam enthalpy at actual outlet conditions (kJ/kg) if steam is used by process;
= saturated water enthalpy at outlet pressure if steam is condensed or exhausted to waste.

Note that if the exhausted steam is used for process, h_{2a}' is equivalent to h_{2a}, and the heat rate is 3,600 kJ/kWh (neglecting gear and generator losses).

To determine the actual heat rate of a turbine to include mechanical losses, the heat rates obtained above are divided by the mechanical efficiency. This actual heat rate is most often used to discribe the performance of a condensing turbine. Performance of a back-pressure turbine exhausting to process is usually described using actual steam rates and turbine efficieny.

Table VI-1

EFFECT OF CHANGES IN OPERATING CONDITIONS ON STEAM RATE

INCREASED PARAMETER	EFFECT ON STEAM RATE
Inlet Pressure	Decrease
Inlet Temperature	Decrease
Outlet Pressure	Increase
Shaft Speed	*Decrease
Number of Stages	Decrease
Leakage	Increase

*Some turbines at certain conditions will actually have an increased steam rate at a higher shaft speed, but generally as shaft speed increases steam rate decreases.

Table VI-2

TYPICAL STEAM RATES AND TURBINE EFFICIENCY FOR TURBINES OF RATED OUTPUT OF LESS THAN 5 MW

TURBINE DESIGN	ACTUAL STEAM RATE kg/kWh	TURBINE EFFICIENCIES %
Back-Pressure Single-Stage	18.0 - 30.0	45 - 65
Back-Pressure Multiple-Stage	6.0 - 22.0	55 - 75
Condensing Multiple-Stage	2.5 - 7.0	65 - 80

In the case of straight-through flow condensing turbines and back-pressure turbines without use of steam exhaust, heat rates provide a useful measure of thermal to electrical energy conversion. Condensing turbines would typically have heat rates ranging from 11,000 to 22,000 kJ/kWh while back-pressure turbines exhausting to waste would have heat rates ranging from 26,000 to 60,000 kJ/kWh. For comparison, back-pressure turbines exhausting to process would typically have a heat rate of 3,900 kJ/kWh, 300 kJ/kWh above the theoretical heat rate of 3,600 kJ/kWh due to gear and generator inefficiencies, as well as miscellaneous heat losses. Heat rates are dependent on the same system features as steam rates and react similarly to changes in these turbine parameters.

Actual heat rates (AHR) in kJ/kWh may be expressed in terms of thermal efficiencies defined as:

$$\eta_{th} = \frac{3,600}{\text{AHR}} \times 100\%$$

Thermal efficiency should not be confused with turbine efficiency, which is a measure only of the design effectiveness. Thermal efficiency allows for all heat and mechanical energy losses of the turbine or the power cycle.

Note that for a back-pressure turbine exhausting all steam to process, the thermal efficiency is in excess of 90%. Typical thermal efficiencies for condensing steam turbines range from 15% to 35%.

Heat rates for organic vapour bottoming cycle turbines range from 26,000 to 60,000 kJ/kWh (Noyes) depending on fluid and operating temperature. These heat rates reflect the low thermal efficiencies associated with these cycles.

Performance characteristics of extraction and induction turbines are generally shown on a graph of throttle steam flow versus power output at various extraction (induction) flows. Such a performance map for an extraction turbine with 4,000 kPa, 400°C inlet steam conditions, 50 mmHg exhaust pressure, and extraction at 1,000 kPa pressure, is shown in Figure VI-3. Steam rates can then be determined by dividing the throttle flow by the power output at the operating point of concern.

Turbines are relatively durable prime movers, requiring minimal maintenance and having long service records. Typical major overhaul schedules run on three to five year intervals depending on size and application. Routine daily equipment checks might include bearing-oil pressure and temperature, gland seal water flow, equipment vibration, and general cleanliness, so as to ensure proper operation of all regulating and safety devices. In the case of

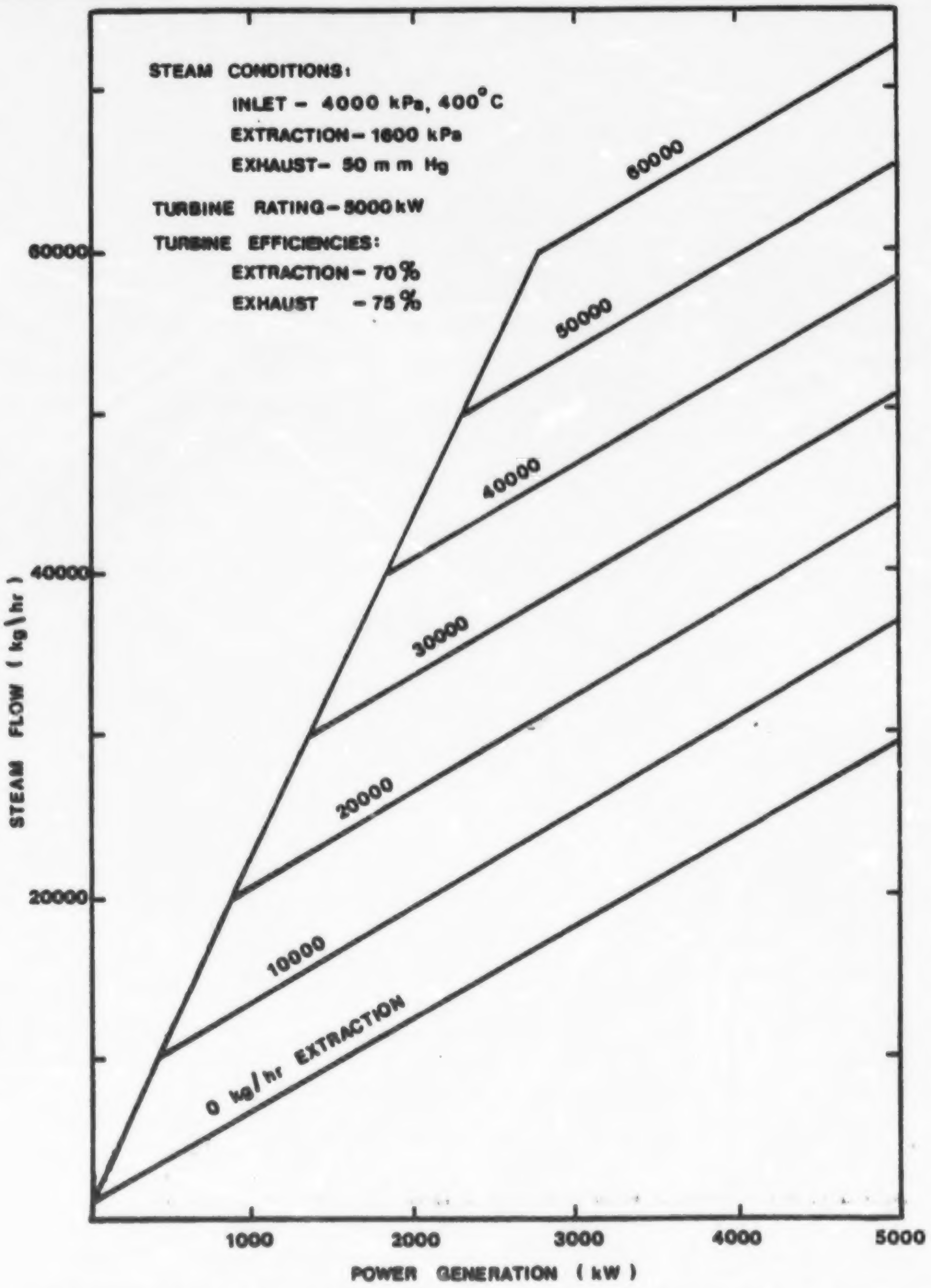


FIGURE VI-3 STEAM FLOW REQUIREMENTS FOR A CONDENSING/EXTRACTION TURBINE

organic vapour turbines, special attention must be made to leaks in the cycle, due to the generally toxic nature of the working fluid.

Costs of straight-through flow turbines are relatively easy to predict because of their simple design compared to extraction turbines. Figure VI-4 (Noyes) shows typical cost estimates for various rated power outputs for back-pressure turbines. The installation costs in this figure include erection, piping, electrical accessories, site preparation, foundations, building, instrumentation and insulation. Installed costs range anywhere from 1.25 to 2.25 times the capital cost (Noyes) and depend on site location and requirements from the above list of items.

Generally, the greatest influence on the dollar per kilowatt equipment cost is the output of the turbine-generator. However, for a specific power output, turbine costs can vary substantially with number of stages, turbine efficiency, and other design parameters. Typically, these per unit output costs range from $150/kW for a larger straight-through flow turbine to over $300/kW for a smaller unit. Extraction and induction turbine costs are more difficult to estimate but a general rule to follow is that they cost approximately twice that of a similarly sized straight-through flow turbine.

The following example (Archibald and Gabriel) illustrates typical costs for installation of a 2,500 kW back-pressure turbine with no extraction and a 5,000 kW condensing/extraction turbine in a forest products mill. The sample mill has a boiler which generates steam at 4,000 kPa and 400°C with a maximum steam flow capacity of 68,000 kg/hr. Process steam requirements are on the average 40,000 kg/hr, with peaks up to 60,000 kg/hr, of steam at 1,700 kPa and 230°C. Taking into account that some power generation will be lost to electrical needs associated with turbine auxiliaries, the actual power outputs based on average process steam requirements will be 1,700 kW and 4,700 kW for the two turbines. Note that the ratio of actual power output to rated power output is much less for the back-pressure turbine than the condensing/extraction turbine since the steam flow through, and therefore the power output of the back-pressure turbine is fixed only by process demands. In the case of the condensing/extraction turbine, steam flow through the entire turbine is independent of process requirements. Typical costs associated with these turbine installations are shown in Table VI-3.

Turbine selection is generally a complicated process due to the number of variables associated with turbine design. The first consideration is the ratio of the process steam (thermal energy) to electrical energy requirements.

Approximate exhaust steam flow versus turbine power output for straight-through back-pressure turbines in the output range of

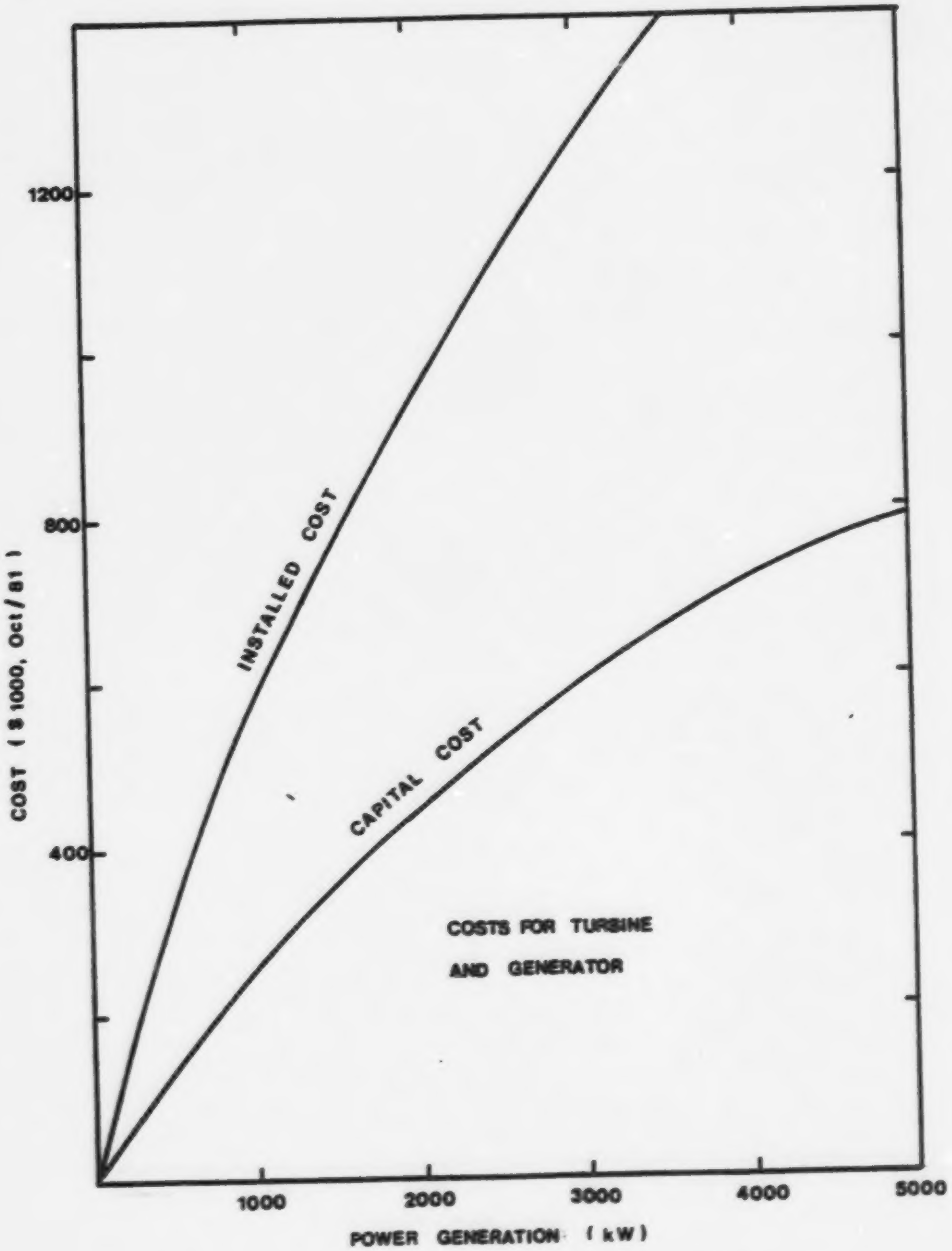


FIGURE VI-4 CAPITAL AND INSTALLED COSTS OF BACK-PRESSURE NON-EXTRACTION TURBINES

Table VI-3

TYPICAL TURBINE CAPITAL AND INSTALLATION COSTS (October, 1981*)

PARAMETER	2,500 kW BACK-PRESSURE TURBINE	5,000 kW CONDENSING/EXTRACTION TURBINE
Average Turbine Net Output	1,700 kW	4,700 kW
Turbine Generator	$ 451,000	$1,115,000
Site Preparation, Foundation, Building	$ 251,000	$ 438,000
Piping, Insulation	$ 251,000	$ 451,000
Instrumentation	$ 63,000	$ 113,000
Electrical and Auxiliaries	$ 276,000	$ 401,000
Total	$1,292,000	$2,518,000
TOTAL Per Net Output	$760/kW	$535/kW
Per Rated Output	$517/kW	$504/kW

* Marshall and Stevens index ratio for 1981/1979 is 1.27.

Source: After Archibald and Gabriel.

interest is shown in Figure VI-5. Should the actual process steam to electricity ratio match those output characteristics of a back-pressure turbine, it would be desirable to use such a turbine if the loads were generally steady.

If the ratio of steam to electrical energy requirements of process were in fact less than those produced by back-pressure turbines, condensing/extraction turbines may be used in a topping cycle. In some cases, the same energy balance can be achieved at a lower cost by using a back-pressure turbine to supply the process steam load and some electrical load and a condensing turbine to make up the remaining electrical load. Alternatively, additional electricity may be purchased from a utility company. If there is appreciable waste heat from process, it may be possible to achieve the same energy balance using a combination of topping and bottoming cycles.

In cases where the process ratio of steam to electrical energy requirements exceeds that of back-pressure turbines, mechanical-drive back-pressure turbines or steam pressure reducing lines may be used to increase the amount of process steam that is available.

In many cases, process energy demands and ratios of thermal to electrical loads vary substantially. Due to their operating flexibility, extraction, induction or combination flow turbines are applied successfully in these situations.

As previously mentioned, turbine and cycle selection can be a complicated procedure, but generally as long as there is sufficient boiler steam pressure and temperature, steam or organic vapour turbine cycles can be fitted to match virtually any steam-electricity load balance (Skratzki).

3. Manufacturers

Several companies were consulted through their head office in Canada, as well as local representatives, for information regarding their steam-turbine generation sets. Of these companies, the following four were selected arbitrarily for presentation of turbine performance characteristics and costs:

a) Worthington (Canada) Ltd. (Turbodyne Steam Turbines)

b) Terry Steam Turbines Company

c) Elliott Steam Turbines

d) Westinghouse Canada Ltd.

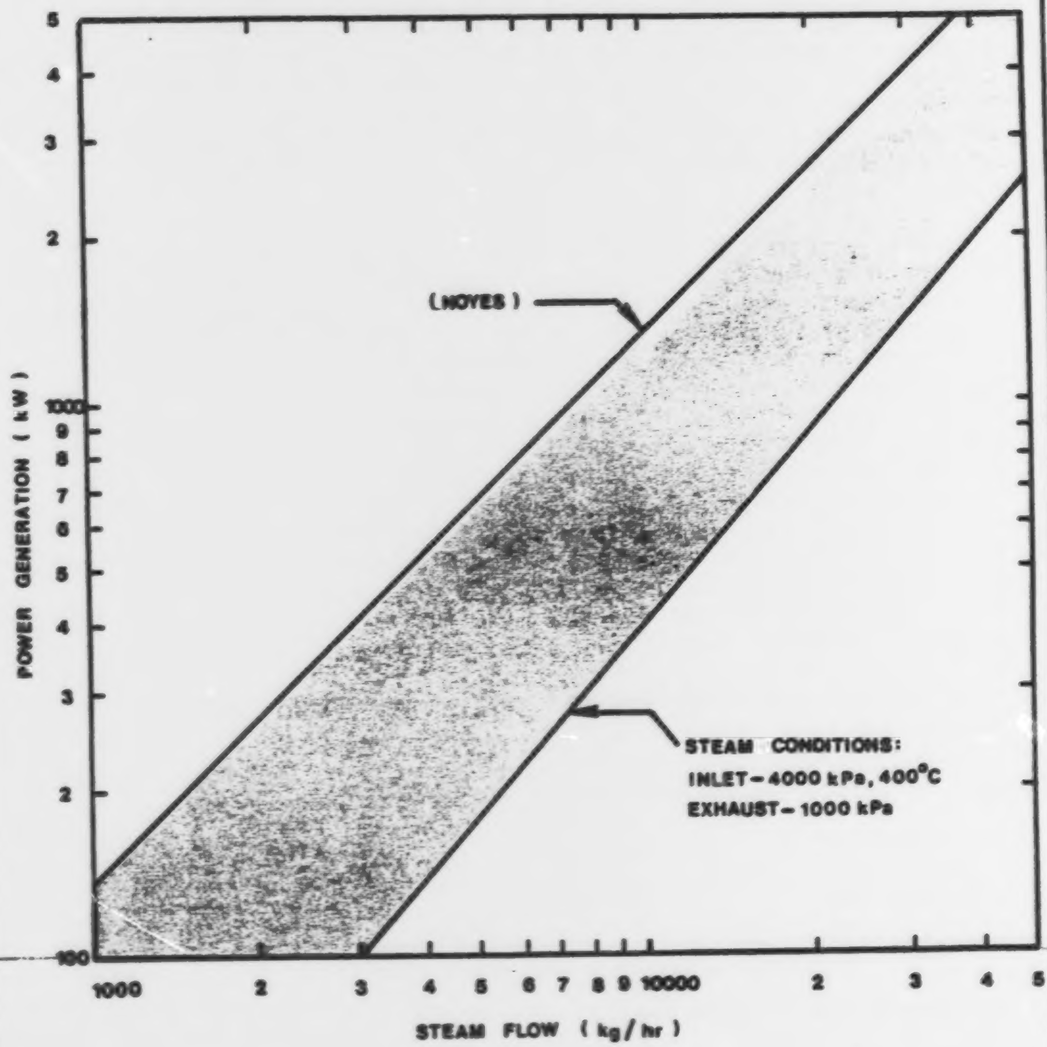


FIGURE VI-5 STEAM FLOW REQUIREMENTS FOR BACK-PRESSURE TURBINES

These four companies are all well established and have had considerable experience in using steam generated from boilers fueled by wood residue. They have various sizes of turbines to provide a complete range of power outputs less than 5 MW and can provide back-pressure or condensing, single- or multiple-staged turbines, with straight-through, extraction or induction flow patterns. In addition, these companies all have the capabilities to provide cycle modifications such as reheat or regenerative cycles.

The selection of operating temperatures and pressures as well as turbine design is dependent on the design of the steam boiler and on process requirements. For the sake of comparison, the steam turbine inlet conditions were chosen as 4,000 kPa and 400°C, common conditions for boilers in forest products mills. Models of steam turbine-generation sets with power outputs of 500, 1,500 and 4,000 kW were selected. Operating characteristics were calculated and budget costs were obtained (Table VI-4) for backpressure systems exhausting at 1,000 kPa and systems condensing at 50 mmHg. The systems considered do not incorporate any cycle modifications such as reheat or regenerative cycles. Extraction or induction flow patterns were not examined due to the complicated nature of their selection and costing. It should also be noted that heat rates calculated for the back-pressure turbines reflect use of exhausted steam by process.

The cost data presented here applies to complete units including:

Turbine

Generator

Governor

Lubrication System

Electrical Control System

Skid Mounting

Other companies that were contacted and submitted information regarding their steam turbine-generator sets include:

AEG-Kanis Turbinenfabrik GMBH

APE Canada Inc.

Coppus Engineering Corp.

Table VI-4

STEAM TURBINE-GENERATOR SETS OPERATING AND COST DATA

COMPANY	RATED[a] OUTPUT (kW)	TYPE[b]	TURBINE SPEED @ RATED OUTPUT (RPM)	ACTUAL STEAM RATE (kg/kWh)	ACTUAL HEAT RATE[c] (kJ/kWh)	TURBINE EFFICIENCY (%)	CAPTIAL COST (1981 $)	COST PER UNIT OUTPUT ($/kW)	DELIVERY TIME (Months)
Worthington (Canada) Ltd.	500	BP/SS	5,000	17.8	3,900	61	130,000	260	9-10
	1,500	BP/SS	5,000	17.6	3,900	62	290,000	193	9-10
	1,500	BP/MS	5,000	16.6	3,900	66	433,000	288	9-10
	1,500	C/MS	6,300	5.3	16,000	64	433,000	288	9-10
	4,000	BP/MS	5,000	14.6	3,900	75	661,000	165	12
	4,000	C/MS	6,300	4.7	14,450	70	661,000	165	12
Terry Steam Turbine Company	500	BP/SS	5,000	17.4	3,900	63	140,000	280	9
	1,500	BP/SS	5,000	17.1	3,900	65	250,000	165	9
	1,500	BP/MS	5,000	16.1	3,900	68	255,000	170	9
	1,500	C/MS	5,000	5.4	16,602	61	245,000	163	9
	4,000	BP/MS	6,000	15.6	3,900	70	---	---	
	4,000	C/MS	6,000	5.0	15,100	67	---	---	
Elliott Steam Turbines	500	BP/SS	5,000	18.4	3,900	60	125,000	250	8
	1,500	BP/SS	5,000	17.3	3,900	63	250,000	166	8
	1,500	BP/MS	6,000	17.0	3,900	64	400,000	267	8
	1,500	C/MS	5,000	5.1	15,500	65	400,000	267	8
	4,000	BP/MS	6,000	16.1	3,900	68	600,000	150	12-14
	4,000	C/MS	6,000	4.7	14,450	71	600,000	150	12-14
Westinghouse Canada Ltd.	500	BP/SS	5,000	18.6	3,900	59	100,000	200	11
	1,500	BP/SS	5,000	18.0	3,900	61	200,000	133	11
	1,500	BP/MS	5,000	16.1	3,900	68	300,000	200	11
	1,500	C/MS	5,000	5.0	15,100	67	330,000	220	11
	4,000	BP/MS	5,000	14.6	3,950	75	600,000	150	13
	4,000	C/MS	5,000	4.5	13,600	74	650,000	163	13

[a] Inlet conditions - 4,000 kPa and 400°C

[b] BP - Back pressure turbine exhausting at 1,000 kPa
C - Condensing turbine exhausting at 50 mmHg
SS - Single-stage
MS - Multiple-stage

[c] Heat rates for back pressure turbines for units exhausting to process, assuming mechanical heat, and generator loss of 7.5%

B. Gas Turbines

1. Cycle Configurations

A gas turbine operates on a Brayton cycle and consists of three components: a compressor, a pressurized combustor and an expansion turbine. Generally, the compressor and turbine are matched for performance and linked by a common shaft. A significant amount of work produced by the turbine must be applied to drive the compressor, thereby reducing overall cycle efficiency.

A simple gas turbine cycle operates by compressing the working fluid from its initial low pressure, increasing the heat content of this compressed air in the combustor, then expanding it through the turbine. The compressor and turbine may either be of axial or radial flow design. Generally the axial flow turbine or compressor is much more elaborate and sophisticated than the radial flow counterpart. As a result, the axial flow unit is more efficient and operates at higher pressures and temperatures. However, the radial-flow machine is more rugged and relatively tolerant to corrosion and erosion. It is also much easier to modify to enable the combustion of low heating-value gas than is the axial flow turbine.

In the past, most gas turbines have burnt fuels in a combustor which is an integral component of the turbine system. With the advent of research into biomass-fueled gas turbines, the main turbine combustor is usually external and independent of the turbine-compressor unit, although generally an integral combustor is maintained to provide start-up and back-up capabilities.

External combustors allow the combustion gases from the combustion of solid or gasified wood to be used as the turbine working fluid. With proper modifications, gases produced from wood gasification and pyrolysis, as well as solid pulverized wood, can be fired in an integral-combustor design gas turbine.

A gas turbine fired with low heating-value gas derived from wood would typically have a total compressor mass flow made up of 85% air and 15% gas, in two compressors. In a regular natural gas-fired turbine the natural gas is already at combustor pressure and only air flows through the compressor. Because of the differences in specific heats of the fuel gas and air, slightly more compressor work is required by the wood gas fired turbine. Hence, net work output is lower, typically by about 5%.

The main drawback of using the "dirty gases" from the combustion of wood as the gas turbine working fluid is their erosive, corrosive and depositional nature with respect to turbine blading and casing. Advances in high temperature heat exchanger technology have led to the investigation of indirectly heated gas turbines to circumvent the problem of turbine wear. In this cycle configuration, clean air is compressed, passed through a heat exchanger where heat is absorbed from the combustion gases and expanded through the turbine.

To further enhance operating performance of the indirectly heated cycle, particularly concerning load following, the working fluid cycle may flow in a closed-loop with the clean gas pre-cooled prior to being compressed.

The turbine cycle may incorporate regeneration, also known as recuperation, in either directly or indirectly heated configurations. Further cycle modifications include intercooling during compression stages and reheating during turbine expansion stages. These modifications, however, are not generally beneficial in the small size range considered in this study.

2. Performance Characteristics

Typically, gas turbine performance is characterized by the turbine inlet temperature, the pressure ratio of the turbine and ambient conditions. A survey of manufacturers by Marksberry and Lindahl showed the maximum allowable continuous inlet temperatures for commercially available gas turbines range from 760°C to 1,050°C (Figure VI-6). Directly heated cycles would provide working fluid temperatures in this region, however, because of heat exchanger limitations, the highest working fluid temperature attainable for indirectly heated cycles would be in the order of 700°C.

Figure VI-7 shows curves of specific work, thermal efficiency, and outlet gas temperature for various turbine inlet temperatures at various turbine pressure ratios for a simple (open) gas turbine cycle. These curves have been generated for ambient conditions of 15°C and 100 kPa with a turbine efficiency of 0.88, a compressor efficiency of 0.83 and a constant pressure loss through the combustor of 15 kPa. Note that the optimum pressure ratio increases as inlet temperature increases. Another point of consideration is that for a constant specific turbine inlet temperature the exhaust gas temperature decreases as the pressure ratio increases. Some process applications require a minimum process supply temperature thereby limiting the pressure ratio of the turbine.

It is generally desirable to run the turbine at high pressures. This effectively decreases the required ducting size and lowers initial costs.

Similar relatively simple calculations can be done to show the effect of ambient temperature and pressure (compressor inlet and turbine exhaust) on specific work and thermal efficiency. Ambient temperature has far more effect on cycle performance than does the

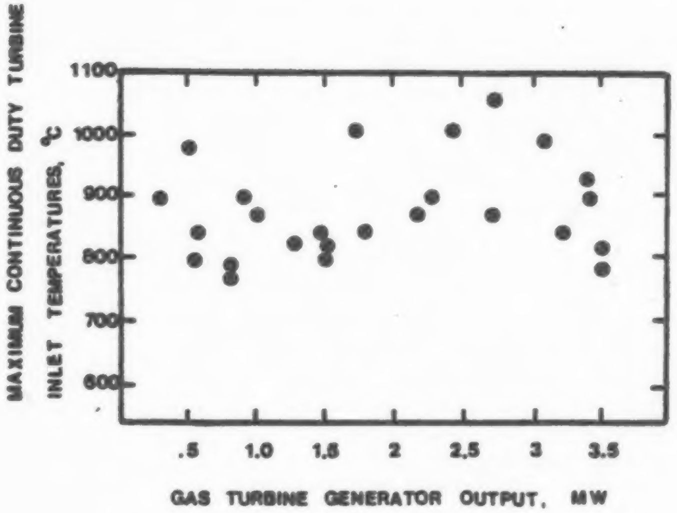


FIGURE VI-6 APPROXIMATE MAXIMUM TURBINE INLET TEMPERATURES FOR INDUSTRIAL GAS TURBINES

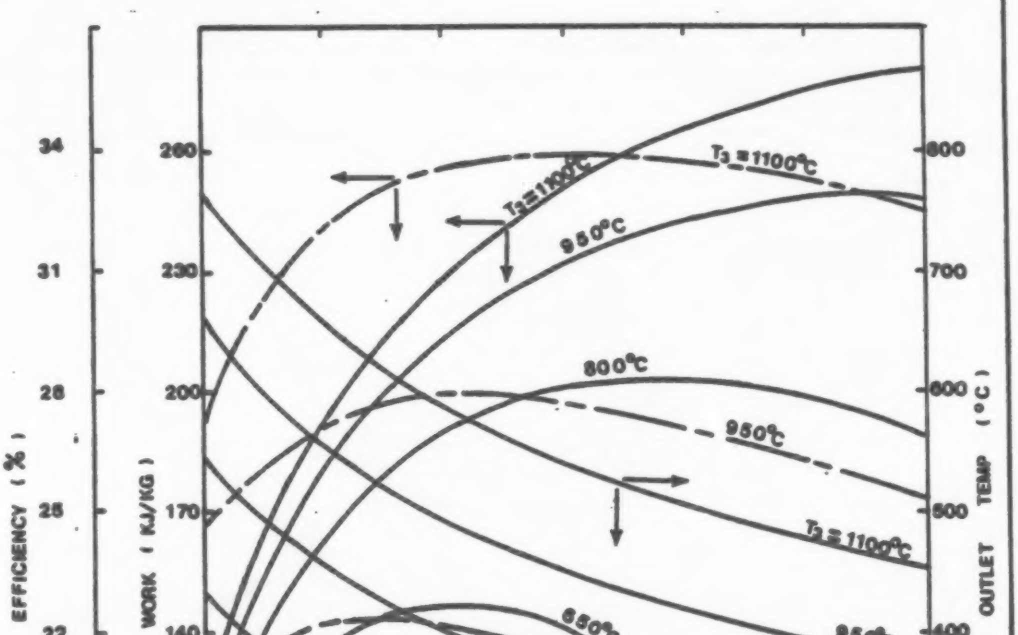


FIGURE VI-7 PERFORMANCE OF SIMPLE CYCLE TURBINE

ambient pressure. An increase in ambient temperature results in a reduction of both specific work and thermal efficiency. For a case with a turbine inlet temperature of 800°C and other factors as per Figure VI-7, increasing the ambient temperature from 15°C to 30°C, results in a reduction of maximum efficiency from 28% to 26%, coincidently with a reduction of specific work from 143 kJ/kg to 131 kJ/kg. In addition, the optimum pressure ratio also decreases with increasing ambient temperature.

The last major consideration of turbine performance is that of mechanical design. Typically, compressor mechanical efficiencies are in the range of 0.8 to 0.85 while turbine efficiencies are slightly higher, usually 0.85 to 0.9. Compressor and turbine efficiencies assumed for calculations shown in Figure VI-7 are within this range.

Varying of compressor and turbine efficiencies results in changes in the thermal efficiency and specific work of the cycle, as well as a shift of the optimal pressure ratio. With a turbine inlet temperature of 800°C and other conditions as stated in Figure VI-7, a reduction in compressor efficiency from 83 to 80% and a reduction of turbine efficiency from 88% to 85%, results in a decrease of maximum thermal efficiency from 28% to 23%, and specific work from 143 kJ/kg to 122 kJ/kg.

By adding a regenerator to the cycle to increase the compressed air temperature prior to its entering the combustor, the cycle performance can be dramatically affected, especially at lower pressure ratios (Figure VI-8). At low pressure ratios thermal efficiencies can be more than double that for similar conditions without regeneration. Also, the trend of exhaust temperature is reversed for regeneration cycles, increasing as pressure ratio increases. It must be pointed out, however, that the benefits of regeneration are minimal if efficient thermal use of the turbine exhaust is achieved using a combined bottoming cycle.

Typically, open gas turbine cycles have heat rates in the order of 5,500 to 6,000 kJ/kWh for systems when the exhausted thermal energy is used by process (Noyes).

Electrical to thermal energy output ratios vary with turbine inlet temperature, pressure ratio and system design (Figure VI-9). These ratios for simple gas turbine cycles typically range from 0.10 to 0.55, which compares favorably to figures obtained by previous studies (0.35 to 0.55 - Fox et al and 0.40 - Jody, Daniels and Bowman).

Electrical to thermal output ratios for regenerative cycles are much higher than for simple gas turbine cycles (Figure VI-10).

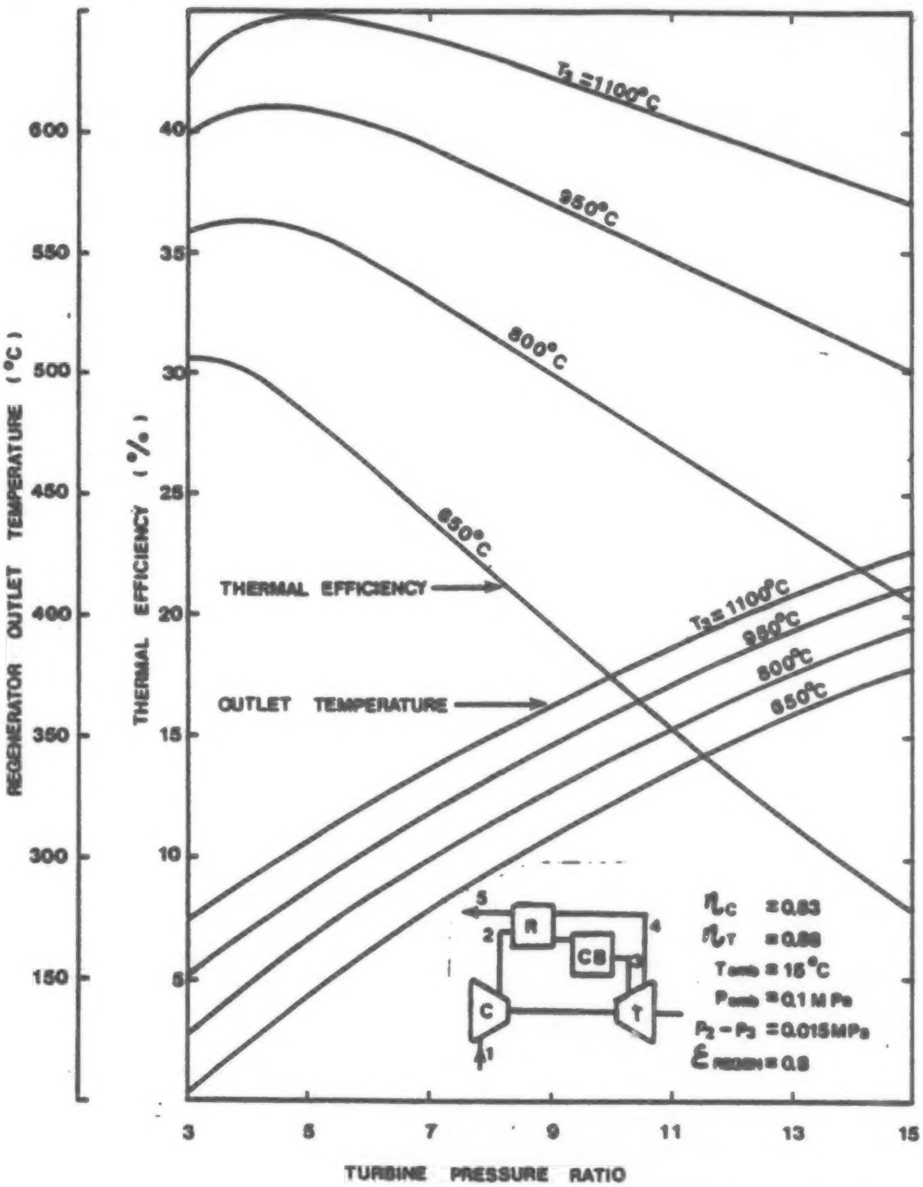


FIGURE VI-8 PERFORMANCE OF REGENERATIVE CYCLE TURBINE

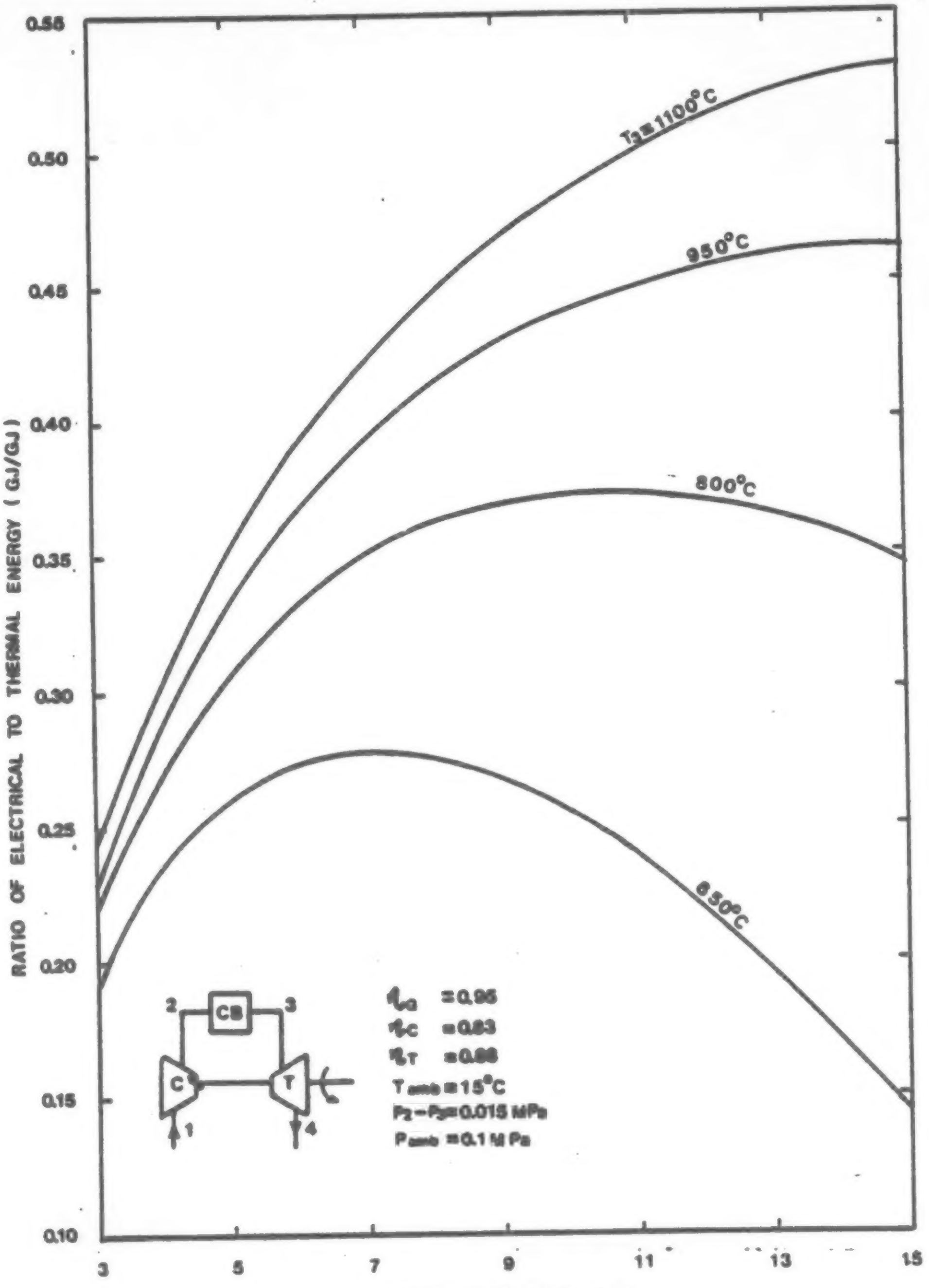


FIGURE VI-9 ELECTRICAL/THERMAL ENERGY FOR SIMPLE CYCLE TURBINE

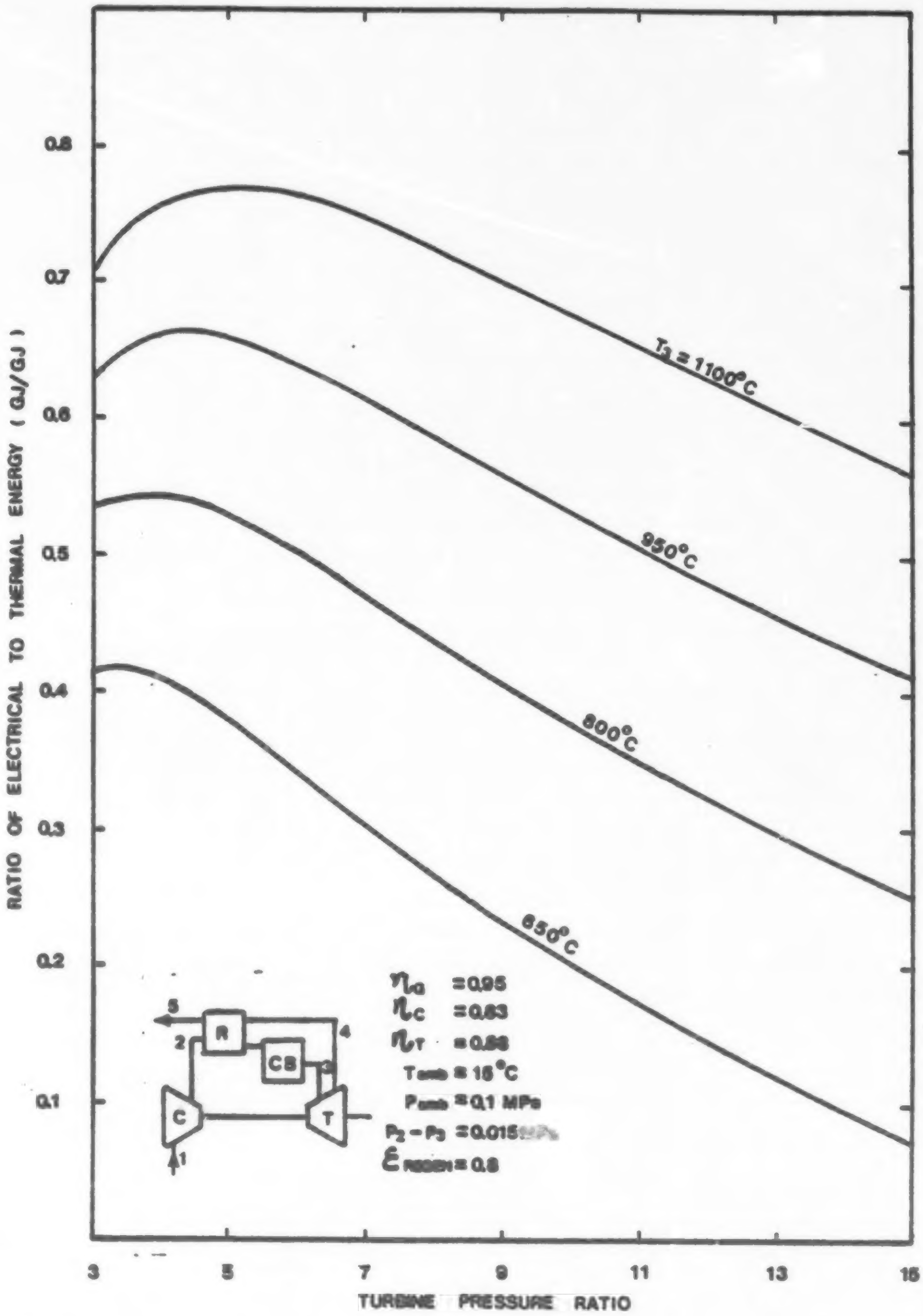


FIGURE VI-10 ELECTRICAL/THERMAL ENERGY FOR REGENERATIVE CYCLE TURBINE

The preceding details of operating performance have been pertinent to all gas turbine cycle designs whether they are directly or indirectly heated, or whether they incorporate open or closed cycles. There are, however, some important differences in the performance of these various cycle designs. One consideration is that at the same conditions, an indirectly heated gas turbine produces about 5% less power than its directly heated counterpart as a result of having no fuel mass flow. Also, the turbine exhaust temperature will be lower for the former because of the difference between thermal dynamic properties of air and combustion gas (Marksberry and Lindahl, 1979).

Another major difference in performance between open and closed gas turbine cycles is that of part-load operation. Part power operation of the open cycle is normally achieved by reducing the turbine inlet temperature. This method has an adverse effect on overall cycle conversion efficiency and decreases the exhaust temperature.

On the other hand, closed cycle gas turbines obtain part-load operation by varying the system pressure (and therefore the mass flow through the turbine) while maintaining the turbine inlet temperature at design conditions. Accordingly, the heat exchanger which recovers heat from the combustor is oversized for the part-load condition and more effective, thus providing maximum cycle efficiency. Also, by increasing the pressure above normal operating conditions, higher than design output can be obtained. Closed cycle gas turbines may be operated from 20% to 130% of full power while obtaining efficiencies equal to or greater than those at full load operation (Harper).

An important difference between the two types of heating is the quality of the working fluid. Whereas air used in indirectly heated cycles is clean, the combustion gases used in directly heated cycles contain ash, char and alkali materials. The alkali material is the major cause of turbine blade deposition and corrosion. It has been found that clay or aluminum silicate additives will react with the alkali material in the wood and greatly reduce its effects (Wocasek and Moody).

The particulate matter is responsible for blade erosion and also contributes to blade deposition. The amount of abrasion and deposition taking place in the expansion section of the turbine depends on the size, shape, hardness and concentration of particulate matter in the gas and, therefore, on the temperature of formation of the particles and the type of hot gas cleanup. The susceptibility of the turbine to wear depends on whether the gas flows axially or radially through the expansion section.

Investigation of wear in turbines due to particulate matter in the inlet gas has been investigated for axial flow designs by the

Westinghouse Research Laboratory. From their study of both previous experience with coal-fired gas turbines and a laboratory turbine with controlled particulate injection, the recommended dust loading distribution given in Table VI-5 was developed. It is evident that only a very small amount of particles greater than 1 micron can be tolerated. These recommendations, of course, apply to particulate from fluid-bed coal combustors, which will differ from particulate from a wood gasifier. The recommended particulate loading in Table VI-5 should be more than adequate for wood gasifier applications.

Maintenance requirements of gas turbines are usually more stringent than for steam turbines. Major system overhauls are required every two to five years, depending on system configuration and design conditions. Usually the combustor requires frequent minor inspections and turbine bearings, rotors, etc. are inspected annually.

3. Manufacturers

Four companies active in the manufacture of gas turbine-generator sets were contacted directly and through local representatives. The companies were:

Detroit Diesel Allison

A/S Kongsberg Vapenfrabrikk

Onan Corporation

Solar Turbines Canada

Costing information of various sizes of turbines manufactured by these companies for traditional fuels may be found in Table IV-6. The prices are for units which include the standard items:

Turbine

Generator

Governor

Fuel System (traditional fuel)

Starting System

Lubricating System

Table VI-5

PROPOSED DUST LOADING DISTRIBUTION IN GAS TURBINE FEED

AVERAGE PARTICLE DIAMETER (um)	WEIGHT PERCENT IN SIZE RANGE	CONCENTRATIONS (g/m^3, 15°C, 101 kPa)
<1	76.8	0.18 - 0.27
1 - 2	16.6	0.056 - 0.07
2 - 3	4.7	0.01 - 0.018
3 - 4	1.6	0.0046 - 0.007
4 - 5	0.2	0.00046 - 0.0007
5 - 6	0.01	0.00023 - 0.00046
6 - 7	0	0

Table VI-6

MANUFACTURER, SIZES, COSTS AND DELIVERY FOR TURBINE/GENERATOR SETS OPERATING ON NATURAL GAS

MANUFACTURER	CONTINUOUS ISO* RATED OUTPUT	UNIT COST		INSTALLED COST		DELIVERY TIME
		($)	($/kW)	($)	($/kW)	(Months)
Detroit Diesel Allison	3,025	1,100,000	364	1,750,000	582	9 - 12
	4,465	1,750,000	392	2,800,000	627	9 - 12
Kongsberg Vapenfabrikk	1,405	600,000	427	960,000	683	6 - 12
	2,700	1,000,000	370	1,600,000	593	6 - 12
Onan Corp.	480	250,000	520	400,000	833	12
	3,940	2,000,000	507	3,200,000	812	12
Solar Turbines Intl.	800	370,000	462	592,000	740	9 - 12
	2,800	1,050,000	375	1,680,000	600	9 - 12
	7,400	3,000,000	405	4,800,000	649	9 - 12

*International Organization for Standardization conditions of 760 mm Hg and 15°C. Reduction of the continuous rated output by 1.1% occurs every 300 m increase in altitude above sea level.

Control System

Skid Frame

Optional equipment available for these units includes:

Sound Attenuating Housing

Water Injection System

Electrical Switch Gear

Inlet Air System

Exhaust Gas System

The continuous rated outputs shown in this table are for conditions of 760 mm Hg and 15°C and are based on natural gas or equivalent fuels with heating values of 37.3 MJ/m^3.

Of the four companies listed, all but Onan Corp. manufacture their own turbines. Onan Corp. packages a complete turbine-generator set using a Garrett gas turbine.

Various models of gas turbines manufactured by these suppliers are summarized in Table IV-7 showing operational characteristics and design information.

a) Solar Turbines

Solar Turbines Canada offers three complete turbine-generator packages, the Saturn (800 kW rated output), the Centaur (2,700 kW rated output), and the Mars (7,400 kW rated output). All packages have axial flow compressors and turbines, and come complete with governor, starting system, lubrication system and instrumentation.

The pressure ratios of these turbines at their rated output range from 6.2 : 1 to 9.4 : 1 and their turbine exhaust temperatures range from 460°C down to 418°C.

Solar Turbines Canda has been involved in developing indirectly heated cycles using their gas turbines, particularly the two smaller units. Currently, the systems are being field tested at 790°C turbine inlet temperatures, but it is hoped that with improved heat exchanger metallurgical technology, this can be eventually raised to 870°C. Exhaust temperatures of these turbines will be just slightly lower than those encountered in the directly heated systems. Solar Turbines Canada requires that particles in the gas steam be

Table VI-7

CHARACTERISTICS OF GAS TURBINE-GENERATORS BY MANUFACTURER

	SOLAR TURBINES CANADA		DETROIT DIESEL ALLISON		ONAN CORPORATION		KONGSBERG VAPENFABRIKK	
MODEL	SATURN	CENTAUR	501 KB	570 KA	560 GTU	4,000 GTU	KG2	KG5
Continuous Output Rating (COR) (kW)	800	2,700	3,025	4,465	480	4,080	1,405	2,700
Nominal Pressure Ratio @ COR	6.2 : 1	9.4 : 1	9.5 : 1	12 : 1	11 : 1		4 : 1	6.3 : 1
Fuel Flow @ COR (kg/hr)	320	967	708	972	448	1,351	766	1,172
Airflow @ COR m^3/sec	7.2	19.6	12.6	15.6	3.0		10.6	17.3
Exhaust Temp. @ COR (°C)	430	441	515	580	470		611	540
Shaft Arrangement	Single	Single	Single	Double	Single	Double	Single	Double
Compressor Arrangement	8-Stage Axial	11-Stage Axial	14-Stage Axial	13-Stage Axial	2-Stage Radial	4-Stage Radial	1-Stage Radial	1-Stage Radial
Turbine Arrangement	3-Stage Axial	3-Stage Axial	4-Stage Axial	4-Stage Axial	3-Stage Axial	3-Stage Axial	1-Stage Radial	1-Stage Axial
Fuel Type	Liquid Gas Dual	Liquid Gas Dual	Liquid Gas Dual	Liquid Gas Dual	Liquid Gas Dual	Liquid Gas Dual	Liquid Gas Dual	Liquid Gas Dual
Supply Pressure For Gaseous Fuel (kPa)	1,140 - 1,380	1,140 - 1,380	2,000	2,000	1,380 - 1,450	1,515 - 1,725	830 - 2,200	1,100 - 2,200

Ratings @ ISO conditions - 15°C, 760 mm Hg at sea level and for fuel with HHV of 42,800 kJ/kg less than 20 microns in size and less than 0.07 g/m^3 in concentration.

b) Detroit Diesel Allison

Detroit Diesel Allison packages three gas turbine generator models - the 501 KB, rated at 3,025 kW; the 570 KA, rated at 4,465 kW; and the 570 KB, rated at 5,060 kW.

The model 501 KB is arranged on a single shaft while the two 570 models have double shaft arrangements, and all packages use axial-flow compressors and turbines. The units come complete with governor, lubrication system, instrumentation and controls, protective devices and necessary accessories for each specific model.

These gas turbine-generator sets run at pressure ratios of between 9.5 : 1 and 12.7 : 1 and exhaust gas at temperatures of 515°C to 580°C at their ISO rated outputs.

Although Detroit Diesel Allison generally require fueling with gas with a heating value of between 33,500 kJ/m^3 and 41,000 kJ/m^3, specialized applications can be considered using gas with heating values as low as 20,500 kJ/m^3. Generally, the gas must be clean (less than 3 microns particle size), dry, free of trace metals and have a low sulphur content.

c) Onan Corporation

Onan Corporation packages two sizes of turbine-generator sets using Garrett Turbines - 480 kW and 4,080 kw. The smaller unit is arranged on a single shaft while the larger is arranged on double shafts. The packages utilize a radial flow compressor and an axial flow turbine. Typically, these units operate at 11 : 1 pressure ratios and exhaust gas at between 450°C and 500°C.

The Garrett Corporation has been involved in the last few years with developing a closed cycle indirectly heated gas turbine. A pilot facility has been considered incorporating a pressurized coal-fired fluid-bed combustor and an 18 kW indirectly-fired, closed-cycle gas turbine. Air is used as the working fluid and provision is made for heat recovery from the bed in the combustor.

The company's main interest since 1974 has been the development of coal-fired fluid-bed combustor systems connected to indirectly-fired gas turbines for utility power plant applications.

d) Kongsberg Vapenfabrikk

Kongsberg Vapenfabrikk manufactures two models of turbines, the KG2 and KG5, which are distributed in North America by the North American Turbine Corporation (NATCO). The turbines are rated at 1,405 kW (KG2) and 2,700 kW (KG5) and both use single stage radial flow compressors. However, whereas the larger unit incorporates an axial flow turbine, the KG2 unit utilizes a more rugged radial flow turbine. The packages operate at relatively low pressure ratios of between 4 : 1 and 6.3 : 1 and exhaust at temperatures of between 540°C and 611°C.

Kongsberg has been actively involved in the design of gas turbines burning low heating value gas. In Wales, a KG2 turbine has been operating successfully since 1978 fueled by mine gas with a heating value of 14,900 kJ/m^3. During the development period of this installation, Kongsberg determined that they could fuel their turbine using gas with a heating value as low as 3,725 kJ/m^3. They also determined that the theoretical limit for gas turbines is 2,200 kJ/m^3.

Modifications required for burning lower heating value gas include minor adjustments to the igniter-plug installation (above 14,900 kJ/m^3) and changes to the gas-fuel nozzle and air swirler (below 14,900 kJ/m^3).

The operation of Kongsberg turbines require that particulate loading of the gas must be less than 30 ppm by weight with a size distribution of:

>5 microns:	maximum 2% by weight
2 - 5 microns:	maximum 18% by weight
0 - 2 microns:	remainder

Proven turbines for operation on wood-derived fuel gas are not presently available, but some development work has been conducted for low heating-value gases. A unit could be developed given that a demand existed. Development of an adequate turbine for low heating-value fuel gas requires an exact definition of the average and the variability of values for the gas composition and heating value, as well as extensive laboratory testing of the turbine under simulated actual operating conditions.

Other manufacturers of gas turbine systems who were contacted and supplied information include:

Alturdyne Corp.

Garrett Corp.

General Electric GT Division

George Engine Co. Inc.

International Systems Inc.

Westinghouse Electric Corporation
  Combustion Turbine Systems Division

A.E.G. Telefunken

C. Internal Combustion Engines

1. Cycle Configurations and Engine Types

In a topping cycle configuration, an internal combustion engine can be used to generate electricity and, with heat recovery from the engine exhaust, steam as well. In fact, two sources of recoverable heat exist for an engine, one being the engine exhaust gases, the other being the engine-cooling medium, either air or water.

Internal combustion engines can be classified into three categories, the fundamental differences between which are the methods of ignition and methods of mixing the air and fuel. The spark-ignited engine operates on an Otto cycle in which combustion takes place at constant volume and ignition is provided by a spark plug. In this cycle the air and fuel is mixed at, or slightly above, stoichiometric proportions in the carburation system prior to injection into the cylinder. Speed and load regulation is attained by throttling the air-fuel mixture (quantitative governing). These engines are fueled by gases or volatile liquids, natural gas being the most common fuel for stationary engine-generator sets. Compression ratios range from 4 : 1 to 12 : 1 (limited by the detonation characteristics of the air-fuel mixture) and may be naturally aspirated or turbocharged.

The second class of internal combustion engine is the compression-ignition engine which operates on a diesel cycle where combustion occurs at constant pressure. Ignition occurs by compressing the air-fuel mixture to the critical detonation pressure of the mixture. Air is supplied to the cylinder unrestricted either by natural aspiration or by turbocharging. Fuel, generally a low volatile liquid, is injected into the cylinder during the compression stroke, at which time the air and fuel are mixed. Speed and load regulation is achieved by throttling the flow of fuel into the cylinders (qualitative governing). This method is limited by the lean limit of flammability and results in low efficiencies at low loads.

In some cases diesel engines operate at sub-critical pressures and ignition is provided by glow plugs. The glow plugs raise the temperature of the air-fuel mixture in the cylinder enough to allow ignition at the lower pressures. Typically, the compression ratios of diesel cycle engines range from 11.5 : 1 to 22 : 1.

The last type of internal combustion engine is the dual-fuel engine. This engine operates on a diesel cycle and is fueled by a high-volatile gas and a low-volatile liquid of good ignition quality which acts as the pilot. The engine operates at compression ratios too low to ignite the primary gas-air mixture, but sufficient to ignite the liquid fuel. The fuel gas and air are mixed in the carburation system prior to injection into the cylinder, and the liquid fuel is injected into the cylinder during the compression stroke. Speed and load regulation of the dual-fuel engine is achieved by quantitative governing of the air-gas fuel mixture or by qualitative governing of the liquid pilot, although the minimum limit of the pilot fuel is approximately 5% of the total heating value of the entire air-fuel mixture.

2. Performance Characteristics

Clean gas produced from a wood gasifier can be used to fuel two of the three types of internal combustion engines, the spark-ignited and the dual fuel.

Spark-ignited engines can be modified from being fueled by natural gas, propane, or gasoline by changing the carburation and injection system. It is possible to burn wood gas fuel using either a naturally-aspirated or a turbocharged engine. However, problems caused by tar deposition on the compressor blades require a suitable gas cleanup system for turbocharged engines. Currently, all engines in actual operation are naturally aspirated, although several companies are in the process of designing turbocharged engines capable of burning wood gas cleaned with cleanup systems now available. Turbocharging increases the capital cost of the engine since two compressors are required, one for the fuel stream and one for the air stream (Figure VI-11). The advantage of turbocharging is that the calorific content per unit volume of the air-fuel mixture is increased and, therefore, the engine output is increased.

Natural gas requires approximately six times as much air for combustion than does wood gas under stoichiometric conditions, but about the same excess air. As a result, the calorific value of the air-fuel mixture of wood gas is approximately 65% to 70% that of

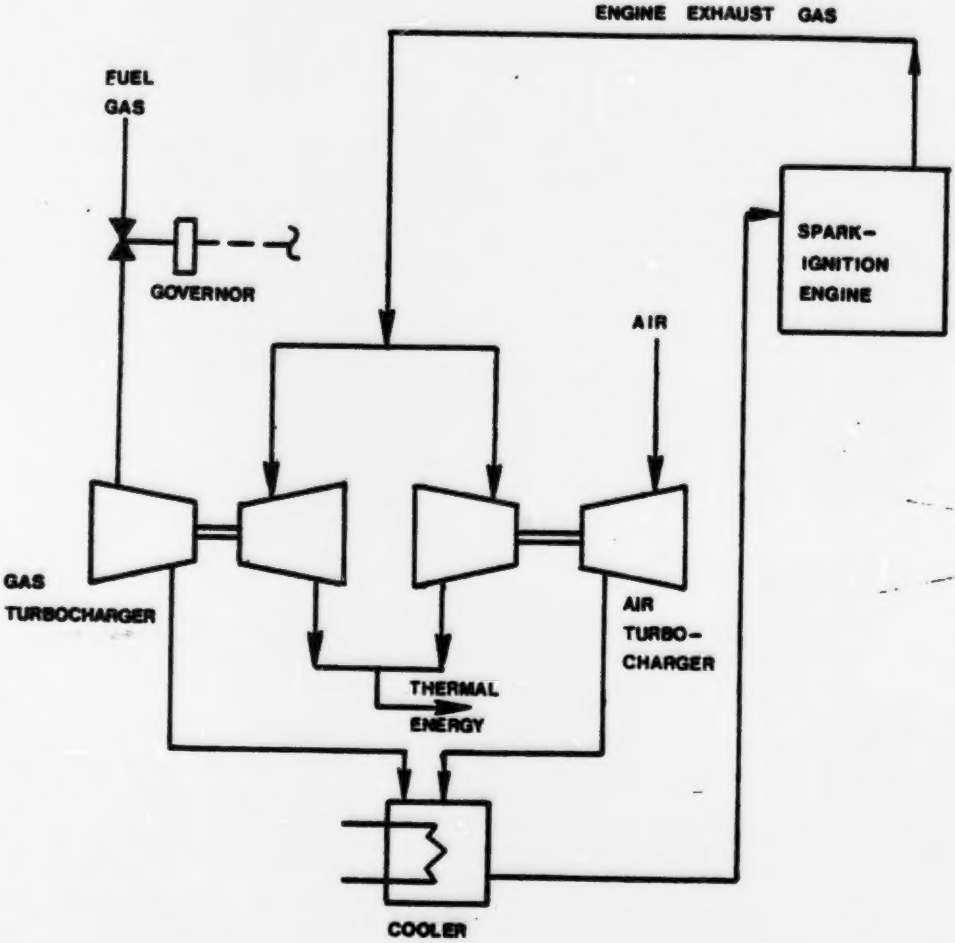


FIGURE VI-11 ARRANGEMENT OF TURBOCHARGED SPARK-IGNITION ENGINE

natural gas. Therefore, engines fueled by wood gas require derating by 30% to 35% from the engine rating on natural gas. Similarly, the de-rating factor from motor gasoline engine ratings is approximately 25% to 30%. These de-rating factors would remain the same whether the engine is turbocharged or naturally aspirated, but would be dependent on the composition of the wood gas.

Thermal efficiencies of spark-ignited engines are generally in the range of 30% to 34% at full load, regardless of the fuel type. Engines are available with turndown ratios of 4 : 1, however efficiences will drop down to a range of 15% to 20% at minimum load conditions. These efficiencies correspond to heat rates of approximately 10,000 to 24,000 kJ/kWh. Spark-ignited engines fueled by wood gas generally have compression ratios ranging from 8 : 1 to 10 : 1, depending on the amount of hydrogen gas present in the wood gas. The greater the amount of hydrogen present in the gas the lower the allowable compression ratio needed to prevent pre-ignition, since hydrogen has a relatively low auto-ignition pressure. Generally, wood gas with a hydrogen content of greater than 30% by volume is not a suitable fuel for spark-ignited engines.

Currently, spark-ignited engines designed specifically to burn low calorific value fuel are available up to 1,500 kW in size. Regular natural gas engines are also available in a similar range of sizes.

Dual-fuel engines are commercially available which can burn a variety of gases, including natural gas, propane and wood gas. In addition, conventional diesel engines may also be modified to enable dual-fuel capabilities. The major difference between diesel and dual-fuel engines is the fuel injection system. There are three methods of injecting the three streams required for combustion. The first of these (Figure VI-12) involves pre-mixing the air and gaseous fuel and injecting this stream and diesel fuel into the engine. The second method involves three separate injections of gaseous fuel, diesel fuel and air into the engine where mixing takes place in the cylinder. The last alternative is to inject three streams consisting of a stoichiometric mixture of gaseous fuel and air, diesel fuel and combustion air for the diesel fuel. In all cases, the gas produced from the gasifier must be cooled and compressed to enhance its volumetric heating value.

As in the case of spark ignition engines, a modified diesel engine will be de-rated when running in the dual-fuel mode because of the lower heating value of the air-fuel mixture and because the dual-fuel engine generally operates at a lower compression ratio (11 : 1 - 14 : 1). The compression ratio is usually limited by detonation characteristics of the fuel gas. Generally, a dual-fuel engine operating on wood gas with a 10% diesel fuel pilot will be de-rated by 20% to 25%, based on the output when on diesel fuel only. This de-rating factor will decrease as the proportion of diesel fuel injection increases.

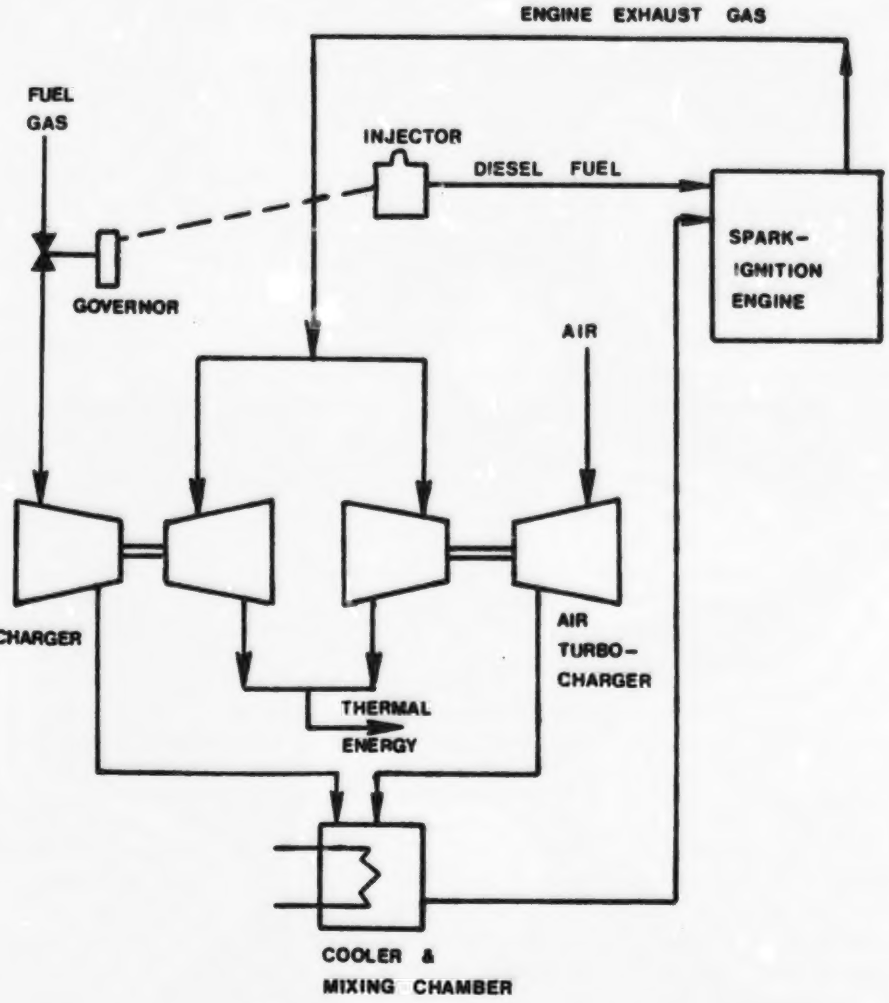


FIGURE VI-12 ARRANGEMENT OF TURBOCHARGED DUAL-FUEL ENGINE

Efficiencies and heat rates of dual-fuel engines are similar to those of spark-ignited engines (Figure VI-13) under full and part load conditions. Diesel engines generally cannot operate over as wide a range as spark-ignited engines, having a turndown ratio of approximately 3 : 1.

Dual-fuel and diesel engines can be obtained in much larger sizes than spark-ignited engines. There are commercially available diesel engines which, when properly modified, can operate as dual-fuel engines with capacities up to 5 MW and larger.

Shaft speeds of both the spark-ignited engine and dual-fuel engine can range from 720 rpm to over 2,500 rpm. Engines operating at lower speeds exhibit less wear and require less maintenance than the higher speed engines, but require larger cylinder displacement for a given output.

Engines require cooling of the block and may either be air- or water-cooled. Air-cooling is generally less costly both initially and during operation, however water-cooling is more conducive to recovery of the thermal energy content. In addition to using thermal energy from the engine-cooling medium, engine exhaust gases may also be used to generate thermal energy. Typically, the exhaust gases have temperatures of between 400°C and 600°C, while the jacket-cooling water is between 75°C and 100°C.

Figure IV-14 shows energy balances for an internal combustion engine operating at full load and at low load (25% to 30% of rated load). The overall efficiency of the engine is 30% to 32% at full load and 17% to 18% at low load. Typically, at full loading, the electrical to thermal energy ratio of internal combustion engines ranges between 0.50 and 0.7, depending on efficiency of the engine. At partial loads the electrical to thermal energy ratios can be related to engine efficiency (Figure IV-15) since this parameter or its specific consumption equivalent is generally specified by manufacturers. Also shown in this figure is the ratio of the electrical output to the thermal content of the exhaust gas only and a specific example of these ratios from data supplied by a manufacturer.

Internal combustion engines are relatively durable engines as long as the fuel gas has been properly cleaned, and the tar and water content has been removed. Routine maintenance includes checking the level and temperature of the lubricating oil and the condition of the cooling system and ignition systems on a daily or weekly basis. Depending on the gas cleanliness and the engine operating conditions, minor overhauls may occur every one to two years and include valve grinds, decarbonizing the engine head, cleaning the heat exchanger, etc.

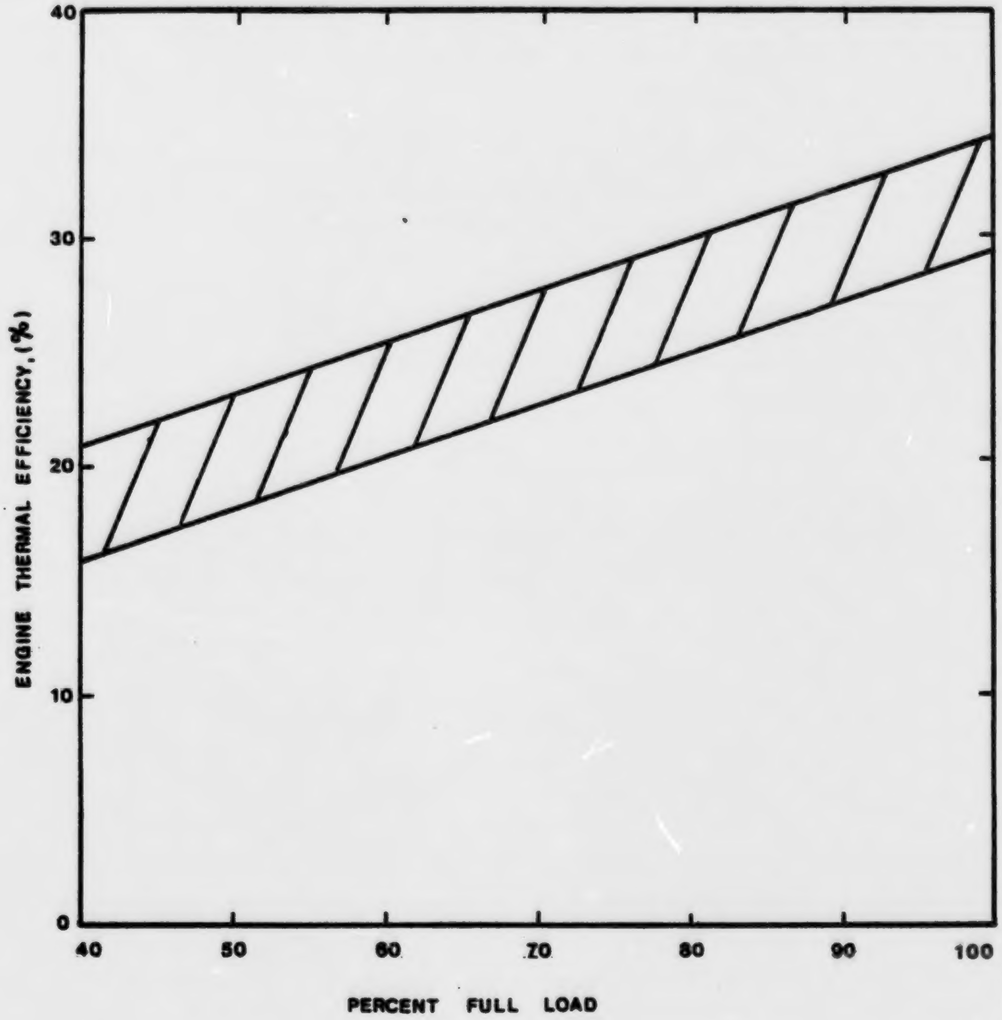
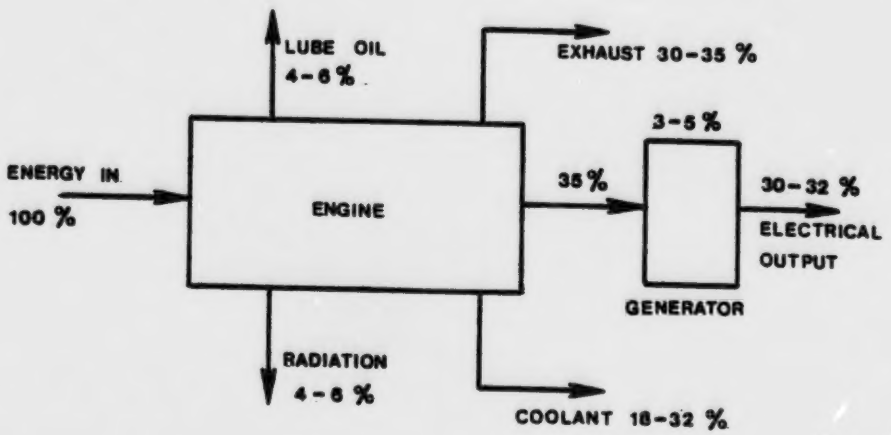
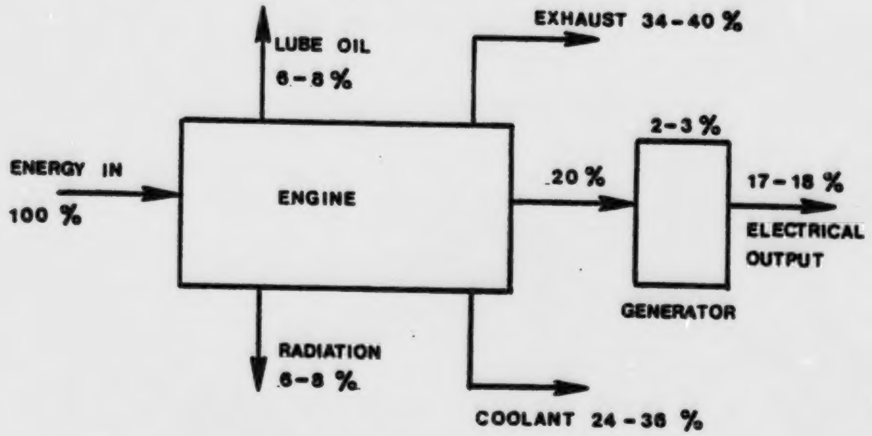


FIGURE VI-13 TYPICAL ENGINE THERMAL EFFICIENCY AT VARIOUS LOADS



A. FULL LOAD



B. LOW LOAD

FIGURE VI-14 ENERGY BALANCE OF INTERNAL COMBUSTION ENGINE

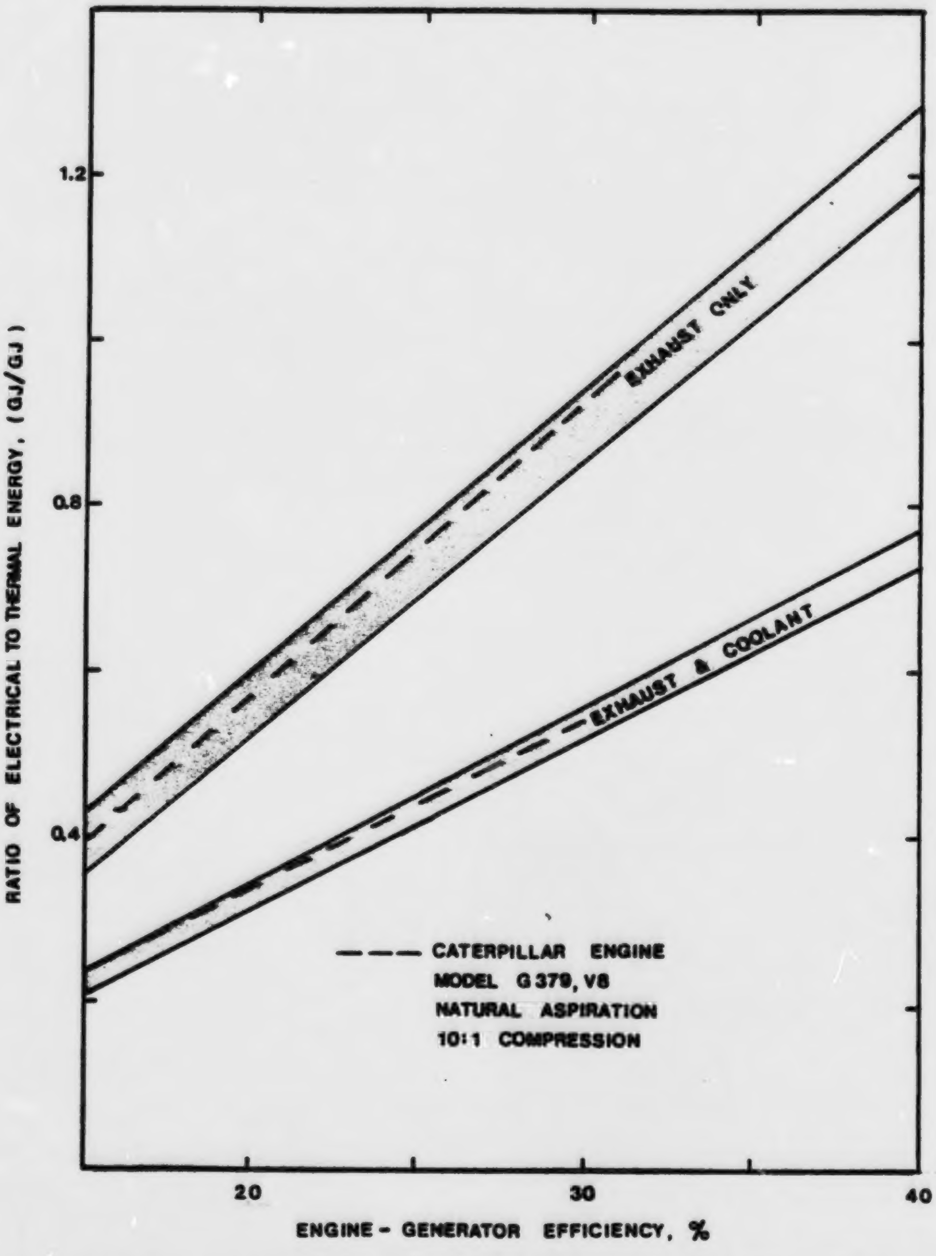


FIGURE VI-15 ELECTRICAL / THERMAL ENERGY FOR INTERNAL COMBUSTION ENGINES

Major overhauls usually occur every five to six years and include stripping the injection pumps, turbocharger, engine and ignition systems and replacing worn out parts.

3. Manufacturers

Several manufacturers of spark-ignited, diesel and dual-fuel engines were contacted and specifications of various models of their products obtained. A list of those contacted and whose typical engine specifications are included in the engine summary table (Table VI-8) include:

Caterpillar

Cummins Rio Grande Sales & Service Ltd.

Deutz Corporation

Moteurs Duvant

MWM Murphy

SACM

In all cases, one or more engine models supplied by these manufactures have been operated using wood gas as the primary fuel. In some cases experience with wood gas fuel is limited, whereas in others the engines are specifically designed for the combustion of low heating value gas.

a) Caterpillar

Caterpillar manufactures both spark-ignited and diesel engines which share common models of engine blocks, i.e. the G379 spark-ignited engine has the identical block of the D379 diesel engine. The spark-ignited engine-generator sets range in outputs of from 85 kW to 650 kW fueled by natural gas and operate with compression ratios of 7 : 1, 7.5 : 1, 8 : 1 and 10 : 1. These engines typically have full load rated efficiencies of 35% operating on natural gas having a lower heating value of 33.74 MJ/m^3.

The diesel-generator sets range in outputs of from 100 kW to 850 kW, based on fueling by diesel fuel with a higher heating value of 45.57 MJ/kg and a density of 848 gram per litre. Typically, compression ratios are 15.1 :1 or 15.5 :1 with engine efficiencies of 35% under full loading.

Table VI-8

SUMMARY OF INTERNAL COMBUSTION ENGINE MANUFACTURER'S SPECIFICATIONS

ENGINE TYPE	SPARK-IGNITED ENGINES[a]									DIESEL ENGINES[b]									DUAL-FUEL ENGINES[c]		
MANUFACTURER	CUMMINS-RIO GRANDE			CATERPILLAR			SACM		CATERPILLAR			DEUTZ DIESEL			MAN MURPHY			SACM		DUVANT
MODEL	G-495	GTA-855	GTA-1710	3306	G379	G399	175 G8SR	240 G8VS	3306	D348	D399	F6L413F	F12L413F	BA8M528	D232V6	TD232V8	TCD232V12	175 G8SR	240 G12V5	6VJG5
OUTPUT (kW)	70	193	416	100	325	650	215	1,200	135	560	850	100	212	982	75	135	250	240	1,480	500
ASPIRATION	Nat.	Turbo.	Turbo.	Nat.	Turbo.	Turbo.	Turbo.	Turbo.	Turbo.	Turbo.	Turbo.	Nat.	Nat.	Turbo.	Nat.	Turbo.	Turbo.	Turbo.	Turbo.	Turbo.
ENGINE SPEED (RPM)	1,800	1,800	1,800	1,500	1,200	1,200	1,000	1,000	1,800	1,800	1,200	2,000	2,300	1,000	1,500	1,500	1,500	1,200	1,200	720
DISPLACEMENT (l)	8.1	14.0	28.0	10.5	32.2	64.4	34.4	204.0	10.5	29.3	64.5	9.6	19.1	85.2	8.8	11.8	17.6	34.4	204	91.9
COMPRESSION RATIO	12:1	8.5:1	10:1	8:1	10:1	10:1	8:1	8:1	15.1:1	16.5:1	15.1:1	18:1	18:1	18:1	16.5:1	16.5:1	14.6:1	11.5:1	11.5:1	13:1
SPECIFIC FUEL CONSUMPTION @ RATED LOAD (MJ/kWh)	11.8	11.6	10.3	10.9	10.5	10.8	10.3	10.3	11.4	11.0	11.8	10.8	11.2	10.7	11.3	10.6	10.8	12.2	11.1	9.2
SPECIFIC FUEL CONSUMPTION @ 50% LOAD (MJ/kWh)	14.6	13.5	12.5	13.3	12.6	12.7	13.6	13.6	12.6	11.8	12.4	---	---	---	---	---	---	15.2	14.8	11.3
COST ($1,000)	231.0	66.1	156.2	32.2	152.0	260.6			32.2	122.0	253.0	25.0	50.0	350.0						402.8
COST PER UNIT OUTPUT ($/kW)	330	342	375	322	468	401			239	218	298	250	236	356						559

a Based on natural gas with LHV = 33.74 MJ/m^3 at ISO conditions at sea level.
b Based on #2 diesel fuels with HHV = 45.57 MJ/kg and density of 848 g/l at ISO conditions at sea level.
c Based on a charge mixture of 90% wood gas LHV = 4.6 MJ/m^3 and 10% diesel fuel HHV = 45.57 MJ/m^3 at ISO conditions at sea level.

Modified spark-ignited versions of the Caterpillar engine have successfully been operated using wood gas. These engines are used in turnkey systems supplied by Biomass Corporation and Pyrenco Ltd. in conjunction with their wood gasification units. De-rating of 25% is required on the engines in this application. Biomass Corporation has plans to test their system with a dual-fuel engine and will likely modify a Caterpillar diesel engine fo this purpose.

b) Cummins Rio Grande Sales & Service Ltd.

Cummins Rio Grande supplies spark-ignited engines ranging in power output of from 65 kW to 450 kW, based on operation using natural gas having a lower heating value of 33.74 MJ/m^3. Engine speeds may range from 900 to 1,800 rpm, although de-rating is required for engine output when running below 1,800 rpm. Compression ratios are set at 8 : 1, 10 : 1 or 12 : 1, and the engines may be naturally aspirated or turbocharged.

c) Deutz Corporation

Deutz Corporation is the North American outlet for KHD Deutz diesel engines manufactured in West Germany. These engines range in outputs of from 2 kW to 7,000 kW, rated on diesel fuel with a higher heating value of 45.57 MJ/kg and a density of 848 g/l. These engines typically have efficiencies ranging from 30% to 34% under full loading and compression ratios of 18 : 1.

d) Moteurs Duvant

Moteurs Duvant manufactures dual-fuel engines specifically for use with their gasification system. Engine outputs range from 85 kW to 740 kW operating at compression ratios of 13 : 1 with a gas having a lower heating value of 4.5 MJ/m^3 produced from their gasifier. The engine may also run in the dual-fuel mode with natural gas replacing the low heating value gas. In this case the engine output is increased by 20% to 35%.

Under full load conditions the engines operate with a diesel pilot of 10% of the total charge of the engine. In the dual-fuel mode, this flow remains constant throughout the range of partial loading. Generally, if the engine is operating at 50% loading or less, the gas produced from the gasification unit is of insufficient quality to fuel the engine, and it must switch from low heating value gas to natural gas or alternatively to a pure diesel operation. In these cases, switching of fuels occurs automatically, however to return to dual-fuel mode with wood gas, manually opeated valves are used.

These systems have been used for several years all over the world. Although these installations have been predominently in Third World countries, the Duvant gasifier-engine-generator system is being considered for installation on Anticosti Island, Quebec, for operation with wood and peat.

e) MWM Murphy

MWM Murphy manufactures diesel engine-generator sets ranging in output from 75 kW to 250 kW. These engines have compression ratios of 14.6 : 1 or 16.5 : 1 and have efficiencies of from 32% to 34% at full load conditions.

An MWM diesel engine has been modified for spark-ignition and is operating in Fort Providence in the Northwest Territories. The engine is fueled by wood gas produced by an Imbert gasifier with a heating value of 4.6 MJ/m^3. Under full load conditions, the engine-generator set produces 120 kW compared to the 150 kW rated when operating on natural gas.

f) SACM

SACM manufactures internal combustion engines specifically for the purpose of burning low heating value wood gas (4.6 MJ/m^3). The cylinder heads are designed so that they may be equipped with either diesel fuel injectors for dual-fuel operations or with electric ignition plugs for straight fuel gas operation. Changing the system from dual-fuel to spark-ignition is rapid and simple and does not require removal of the head or rocker system.

The engines are available in outputs of from 150 kW to 1,500 kW at full load conditions operating on wood gas. If natural gas is used in place of wood gas, the maximum output can be raised to 1,800 kW. The engine operates at a compression ratio of 8 : 1 in the spark-ignition mode and 12 : 1 in the dual-fuel mode. Engine efficiencies of the spark-ignited version are approximately 35%, while those of the dual-fuel mode range from 30% to 34%.

D. Stirling Engines

1. Cycle Configurations

The Stirling engine has been in existence since 1816 and was used extensively in Europe and the United States. These early Stirling engines, or hot air engines as they were then known, were unsophisticated and operated at low efficiencies, and their use was eventually abandoned with the development of the spark-ignited and diesel engine.

In 1937, N.V. Philips Research Laboratories of the Netherlands resurrected the concept of the Stirling engine and have brought it to a relatively advanced state. At present, this company is concentrating its efforts around employing the Stirling cycle as a cryogenic cooling engine, currently the only established application of this cycle. Several companies have obtained licences from Philips to use their technology to develop a suitable Stirling engine for applications in the automotive and electrical power generation fields. These include United Stirling of Sweden and its American subsidiary Stirling Power, the Ford Motor Company, Entwicklungsgruppe Stirling Motor M.A.N. and MWM of Germany.

The major components of a simple Stirling engine are of an expansion cylinder and piston, a heater, a regenerator, a cooler and a compression cylinder and piston (Figure VI-16). In the Stirling engine the working gas, usually helium or hydrogen, operates in a closed cycle moving back and forth between the expansion and compression cylinders. Starting at the compression space, the cooled gas is compressed, passed through the cooler, then through the regenerator where heat is added, then through the heater where its temperature is brought up to operating conditions and finally expanded in the expansion cylinder. The gas then returns through the heater and regenerator where its heat content is partially taken up by the regenerator and through the cooler where its temperature is decreased prior to entering the compression cylinder.

The ideal Stirling cycle consists of isothermal compression and expansion at two separate temperatures. Because of this and the inclusion of regeneration, the Stirling cycle efficiency in theory should be the same as that of a Carnot cycle operating between the same temperatures. The thermal efficiency of a Carnot cycle, and hence the Stirling cycle, operating between 20°C (293 K) and 800°C (1,073 K) should be approximately 73%. In current research and development this case has never been obtained because of mechanical friction, pumping losses and other heat losses.

The process of compression, heating, expansion and cooling occur simultaneously, but partially overlap in time in the Stirling

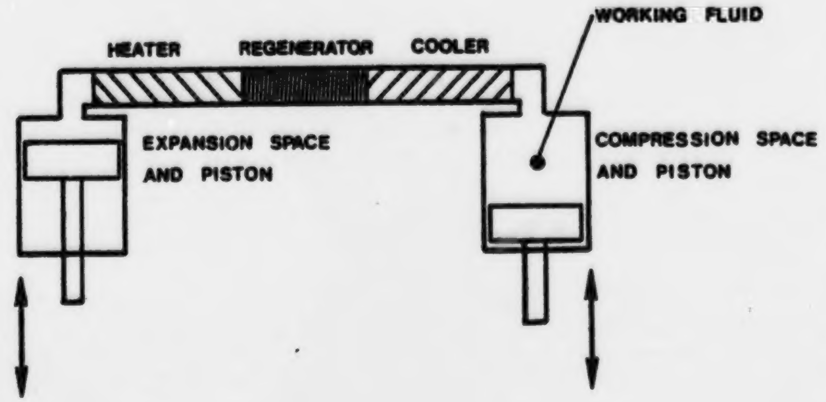


FIGURE VI-16 COMPONENTS OF A SIMPLE STIRLING ENGINE

engine. In addition, although the processes occur in different parts of the engine, their boundaries are blurred. Therefore, unlike other engines in which these processes can be well defined by time or spatial separation, it is extremely difficult to calculate, with any degree of confidence, the complex processes occurring inside a practical Stirling engine (Collie).

Two approaches have been proposed for the practical application of a Stirling engine fueled by biomass. Similar in concept to the gas turbine, these approaches are the integral and external combustor systems. With the integral heater concept, the combustion of wood-derived fuel gas occurs in the heater section, which is a fixed part of the engine. With an externally fired Stirling engine, wood is burned in an external combustor and the hot combustion products are passed through the heater of the engine. In both cases thermal energy is recovered from the engine's cooling system, usually water, and the exhaust gases.

2. Operating Characteristics

At the present time, there are no Stirling engine sizes greater than 40 kW output which are commercially available. Currently there is a great deal of research and development of Stirling engines in the 200 to 300 kW size range and some on engines with up to 2,500 kW output. Most of the work currently underway involves computer modeling and analysis with the hopes of obtaining a detailed understanding of the complex processes occurring inside a Stirling engine. Unfortunately, because of the licencing practices concerning the Stirling engine design, most of the existing operating data are proprietary and difficult to obtain.

Some of the available data, which includes predicted values for larger engines and actual values for smaller engines, are shown in Table VI-9.

There are several attractive features of the Stirling engine, the most important of which is its high thermal efficiency possibilities. As previously stated, the Stirling engine in theory should operate with the Carnot efficiency. However, because of inherent mechanical and thermal losses in the engine, this Carnot efficiency cannot be achieved. It has been suggested that currently, the Stirling engine can obtain 65% to 70% of the Carnot efficiency and that with improved engine design and heat exchanger technology, an overall thermal efficiency of 60% can be achieved (Percival). From Table VI-9 it is seen that currently the maximum attainable thermal efficiency is in the range of 35% which corresponds to approximately 50% of the Carnot efficiency in that temperature range.

Table VI-9

PREDICTED AND ACTUAL TEST RESULTS OF VARIOUS STIRLING CYCLE ENGINES

SOURCE	Toscano et al	Uherka et al	Richards & Lehrfeld	Stirling Power Systems V160 Johansson & Agnu	United Stirling 4-95 Bratt
Status	Predicted	Predicted	Actual Test	Actual Test	Actual Test
Heat Source	Clean Fuel	Hot Gas	Electricity	Liquid Fuel	Liquid Fuel
Heater Temp. (°C)	720	720-800	750	650	720
Cooler Temp. (°C)	50-80	70-80	50		50
Max. Power Output @ Generator (kW)	373-2,237	373	1.0	6.5	30
Max. Electrical Generating Efficiency (%)	25-34	28-29	28.6	22	15-33
Electrical to Thermal Energy Ratio		0.52-0.56		0.31	
Engine Speed (rpm)	1,200	900-1,200	1,950	1,800	1,000-4,000
Operating Gas and Pressure (MPa)	Hydrogen or Helium 15	Helium 10-15	Helium 9.9	Helium 5-6	Hydrogen 3.5-15
Engine Cost ($/kW)	224 (1980)	337-463 (1980)			400-500 (1980)

The thermal efficiency of a Stirling engine depends primarily on the temperature of the heater (heat source) and the cooler (heat sink). Other influencing variables are the pressure of the working fluid and the engine speed. Figure VI-17 depicts the performance curves of the P40 engine, manufactured by United Stirling, showing the effect of these parameters on thermal efficiency and power output (Bratt).

It has been demonstrated that the Stirling engine can operate down to 20% of its rated output with relatively little loss of thermal efficiency (Figure VI-18 [Neal]). This figure also shows another significant feature of the Stirling engine, its high total energy recovery capabilities. Because of the engine design, up to 90% thermal efficiencies can be realized if the recoverable thermal energy is properly utilized. This compares to 70% to 75% thermal efficiency of a back-pressure turbine and internal combustion engines in the same size range.

In the size range currently proposed for Stirling engines, less than 2,500 kW, they will be in direct competition with internal combustion engines. With current technology, the Stirling engine already has superior heat rates (and therefore efficiencies) over the entire range of operating loads compared to internal combustion engines (Walker).

Other advantages of the Stirling engine over internal combustion engines include lower emissions and noise levels. Typically, the Stirling engine exhibits hydrocarbon and carbon monoxide emission levels which are much less (by an order of magnitude) than Otto and Diesel engines and oxides of nitrogen emissions comparable to these other engines. The Stirling engine typically operates at a noise level of around 70 decibels compared to 85 to 95 decibels of internal combustion engines.

Controls and instrumentation are minimal in a Stirling engine. Modulation of the fuel and air supply is required to control the combustor-heater temperature. The engine loading is controlled by the working gas pressure which is usually achieved by a by-pass and reservoir system running off the compression cylinder.

Since combustion is external of the cylinders, maintenance is generally minimal. Routine maintenance would include inspection of the combustor and pressure control systems and checking of the engine lubrication system. Ignitors would be replaced and the working gas level topped every four to six months.

3. Manufacturers

There are currently two manufacturers that have Stirling engines that could be termed commercially or near-commercially

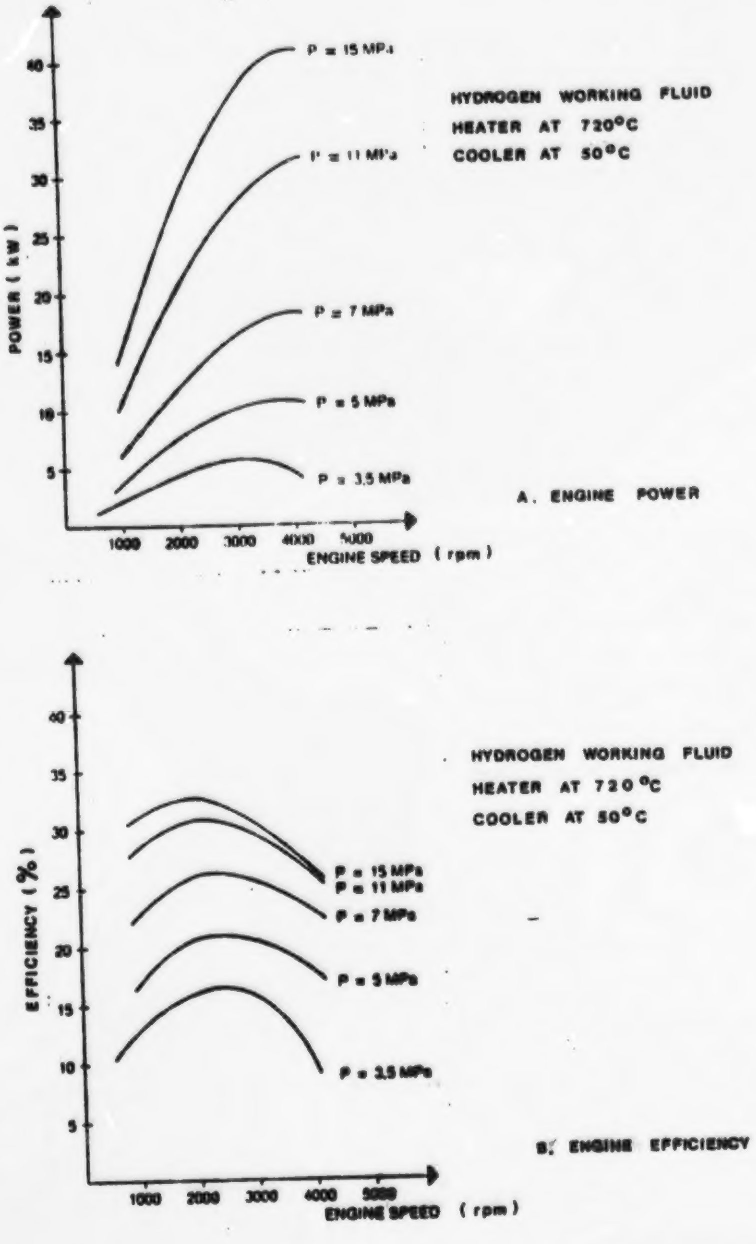


FIGURE VI-17 POWER AND EFFICIENCY CURVES FOR STIRLING ENGINE

SOURCE: BRATT

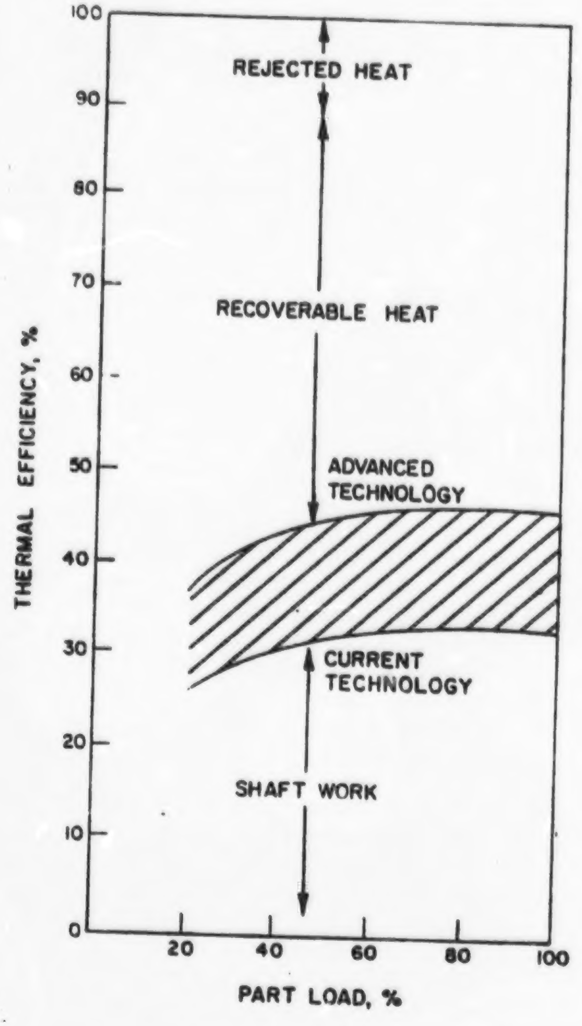


FIGURE VI-18 STIRLING ENGINE THERMAL EFFICIENCY

SOURCE: NEAL

available. These are United Stirling of Malmo, Sweden and Stirling Power Systems of the United States. Although these companies are completely separate with different marketing approaches, they are linked by the company FFV of Sweden who owns interests in both companies.

Although the technology of both companies is identical, they differ in that the Swedish company concentrates on large engines up to 150 kW, while the American company is involved with smaller engines less than 15 kW.

a) United Stirling

United Stirling was formed in 1968 as a joint venture company by two Swedish firms, FFV and Kockums. A license was obtained from N.V. Philips Company and since that time, work has progressed in the development of the Stirling cycle engine.

Originally, this company concentrated for the most part on short range automotive applications in the size range of 40 kW to 150 kW shaft output. Two engines were designed and have been tested thus far, the 4-95 and the 4-275 with shaft outputs of 40 and 75 kW, respectively, at an engine speed of 4,000 rpm. Typical performance curves for the 4-95 engine are shown in the previous section in Figure VI-17 (Bratt). Testing on these engines has shown that thermal efficiencies, of the engine only, in the order of 39% can be achieved.

These engines both utilize a four cylinder double-acting piston arrangement and can operate with either hydrogen or helium as the working fluid. In addition to these two engines, an engine with a shaft output of 150 kW has also been designed. This engine consists of two 4-275 engine modules connected by a common crankcase.

In 1979, United Stirling, Inc. was established in the United States by United Stirling as a wholly owned subsidiary. This company, with technical guidance from its parent company, has been contracted by the U.S. Department of Energy under the supervision of Batelle Pacific Northwest Laboratories to investigate the direct and indirect combustion of wood combined with a Stirling engine (Gummesson, Hansson & Percival). The indirect combustion method involves gasifying wood and burning this gas in an external cyclonic combustor. This program has advanced to the point where an adequate fuel metering system has been developed, and a suitable combustor arrangement has been designed and tested.

The next phase of this program, the results of which should be available in the summer of 1983, covers the design and testing of this system connected to a Model 4-95 engine. It is hoped that at least 100 hours of engine operation will be achieved at the design shaft output of 25 kW. From preliminary investigations, it is expected that the overall thermal efficiencies of this system should be in the order of 25% to 30%. A commercial system with an optimized gasifer-Stirling engine should have a greater efficiency than for this experimental system. The company hopes if all goes well that commercialization of scaled-up systems could be as near as three years in the future.

At this time, this engine could be purchased from United Stirling. Typically, these engines would cost between 400 and 600 $/kW, depending on system configuration.

In addition to this current program and other ongoing testing programs, United Stirling together with Advanced Mechanical Technology, Inc. and Thermacore, Inc of the United States, have been involved with a conceptual design of a larger 400 kW Stirling engine coupled with a coal fluid-bed combustor (Schultz et al). The concept of this engine was similar to models already developed using helium as the working gas and operating between 720°C and 80°C and at a mean pressure of 15 MPa. This engine would have a shaft output of 452 kW at 38% efficiency and operate at 900 rpm. The coal-pile-to-busbar electric conversion efficiency is estimated to be in the order of 25% and with proper utilization of waste heat, a total thermal efficiency of 80% could be expected.

It was estimated that the total cost of these 400 kW system modules, including fluid-bed combustor, engine- generator, controls and instrumentation, between $1,000 and $1,500 per kW installed. System configurations of this kind could be demonstrated within five years and commercialized by 1990.

b) Stirling Power Systems

In the mid 1970's, FFV, formed a joint venture company with Thetford Corporation of the United States called Stirling Power Systems. This company's main activity has been involved with commercial development of small Stirling engines with outputs of 15 kW or less. One of their primary markets has been the recreational mobile home vehicle for which they could supply engines that provide electrical power, ventilation and air conditioning.

This engine, the V160, operates with a low noise level and requires little volume, its installation dimensions being 700 mm by 780 mm by 900 mm. This engine produces 15 kW of power at the shaft, which is converted to approximately 7 kW electrical power. The difference in shaft and electrical power is taken up by engine auxiliaries and in providing the air conditioning and ventilation services. Speed control, power control and air-fuel control are automatically operated and self-adjusted.

Typically, the engine operates with a moderate heater temperature of 650°C to 700°C, increasing its durability. The engine can operate between speeds of 600 rpm to 3,000 rpm, but typically operates between 1,500 rpm to 1,800 rpm. At these speeds and at the operating gas pressure of 15 MPa, the energy thermal efficiency is approximately 35%.

E. Steam Engines

1. Cycle Configurations

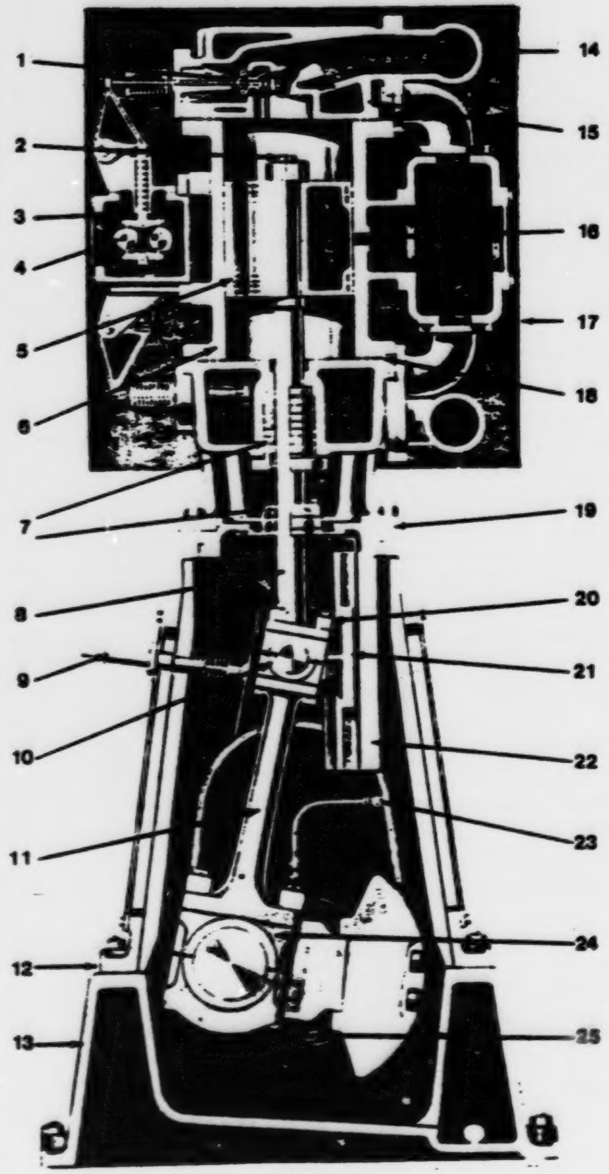
The steam engine was the original steam driven prime mover developed by James Watt. It so remained unchanged until the 1930's when the steam turbine replaced it in most applications and since that time, development of this prime mover has remained relatively dormant.

The steam engine cycle consists of expanding high pressure steam within one or more cylinders of a steam engine, producing a reciprocating piston action. This action is translated into a rotary motion through a crankshaft connection (Figure VI-19).

For the most part, the steam engine cycle configurations follow those of the steam turbine. The engine may exhaust steam at some back-pressure which is used for process thermal energy or may be of the condensing kind, either with or without steam extraction. In the case where steam is extracted from the engine only, multiple-cylinder engines can be used where the cylinders are connected in series.

An important recent development in steam engine design is that of the modular concept of the steam motor of Spilling-Werk A.G. of West Germany. In this design, basic engine units of one cylinder each can be interconnected to form multiple-cylinder engines. This concept is discussed in more detail under Manufacturers of this section.

Another development currently being considered is a rotary action steam engine. This concept is still in its early stages as far as steam engines are concerned, and is being studied by the Westinghouse Electric Corporation (Tauson).



1 Double-seat steam admission valve (each end of cylinder)
2 Piston rod lock
3 Pressurized lubrication to all cam mechanisms
4 Dual camshafts
5 Piston
6 Cylinder
7 Steam packing (top) Bulkhead packing (bottom)
8 Piston rod
9 Indicator reducing motion for each cylinder
10 Crosshead and pins
11 Connecting rod
12 Totally enclosed frame
13 Engine base
14 Inlet steam piping
15 Cylinder head
16 Exhaust port and manifold
17 Cylinder jacket
18 Double ground joint
19 Accurate cylinder - frame alignment
20 Crosshead bolts
21 Crosshead shoe
22 Crosshead guide
23 Pressurized lubrication to all reciprocating parts
24 Connecting rod bearing box
25 Main bearing shell

FIGURE VI-19 SKINNER STEAM ENGINE

2. Operating Characteristics

During the period in which the steam engine was widely used, the engines were built in sizes ranging from childrens' toys to 19,000 kW (Marks). The modern steam engine or motor is now available in outputs ranging from 25 to 2,000 kW. Steam engines can be operated using steam at pressures from 400 to 15,000 kPa, however, steam turbines are more economical in the larger sizes and thus, steam engines are generally limited to applications of small output using lower pressures.

Table VI-10 shows the comparison of steam turbines and engines at the same steam conditions and flow rates (SNC).

Performance of steam engines is indicated with the same terms as used for steam turbines. Typically, engine efficiencies, including generator, range from 0.5 to 0.85. As with turbines, back-pressure engines are characterized by their steam rates which range from 11 kg/kWh to 28 kg/kWh, depending on steam conditions at the inlet and exhaust and engine efficiences (Figure VI-20). Steam rates for condensing engines are also often given and may range from 5 kg/kWh to 11 kg/kWh. More often, however, condensing engines are characterized by their thermal efficiencies which range from 14% to 24%, and which also depend on steam conditions and engine design.

As with turbines, exhaust steam from back-pressure steam engines is suitable for thermal process applications. In these cases thermal efficiencies of the overall systems reach well over 80%. Electric to thermal energy ratios for these back-pressure engines can range from 0.06 to 0.16 (Figure VI-21).

When a steam engine is used to drive a generator, it is important to maintain a constant shaft speed throughout one revolution of the crank. To achieve this, a heavy flywheel is generally used. To maintain shaft speed during load change, a throttling governor is used to control the throttle valve in the main steam line to the engine. The governor is generally a fly ball type, attached to the engine shaft, which responds to variations in engine speed and activates the throttle valve. This governor also acts as an overspeed trip device. Generally, an oil hydraulic control is often employed to regulate speed range within small limits.

For the most part, steam engine performance can be compared to that of the steam turbine. Advantages and disadvantages of the application of the steam engine versus the steam turbine are as follows:

Table VI-10

APPROXIMATE POWER (kW) FROM RECIPROCATING ENGINES (E) AND TURBINES (T)
PASSING 4,537 kg OF DRY STEAM PER HOUR AT VARIOUS INLET AND EXHAUST PRESSURES

EXHAUST PRESSURE (kPa)	32	49	66	84	101	136	170	239	308	277	446	515	584	653	791	936
INLET TEMP. °C, DRY	72	82	89	95	100	109	116	126	134	142	148	153	158	162	170	178
INLET PRESSURE (kPa):																
690 E.	328	317	306	294	283	246	209	168	130	97	-	-	-	-	-	-
690 T.	429	350	313	272	235	197	164	123	93	63	37	-	-	-	-	-
862 E.	362	350	335	324	313	283	246	194	169	130	104	-	-	-	-	-
862 T.	436	365	328	291	253	220	186	145	112	82	60	37	-	-	-	-
1034 E.	376	369	350	339	328	306	276	231	186	157	127	108	93	-	-	-
1034 T.	447	380	343	313	276	246	212	168	130	104	78	60	37	-	-	-
1206 E.	395	380	365	354	343	321	298	253	209	175	153	134	112	97	-	-
1206 T.	458	395	354	324	291	257	224	183	149	123	97	78	60	-	-	-
1380 E.	403	388	373	362	350	332	313	272	239	205	175	153	134	115	89	-
1380 T.	473	406	369	335	306	268	235	197	168	142	119	97	82-	67-	-	-
1550 E.	410	395	380	369	358	339	321	283	253	227	197	168	153	134	104	78
1550 T.	485	417	380	350	321	287	246	212	183	153	130	104	93	78	-	-
1724 E.	-	-	-	-	365	345	328	291	268	242	212	186	168	149	119	89
1724 T.	492	429	391	362	332	298	261	224	194	164	142	119	104	89	-	-
2070 E.	-	-	-	-	-	-	343	313	287	265	242	212	190	168	142	115
2070 T.	503	447	410	380	354	321	291	246	212	183	160	134	119	101	-	-
2413 E.	-	-	-	-	-	-	358	328	302	280	257	231	205	186	160	134
2413 T.	522	466	432	395	369	335	306	257	227	198	175	153	138	119	-	-
2760 E.	-	-	-	-	-	-	-	343	321	298	276	246	224	201	175	149
2760 T.	537	485	447	410	380	350	317	272	239	212	190	171	157	134	104	-

Note: Approximate increase of power by superheatng: 55.5°C: 17 1/2 per cent; 83°C: 25 per cent.
The power figures apply to one unit requiring the amount of steam stated at full load, the power plant being of a type and size best suited to the particular inlet and exhaust conditions.

Source: After SNC

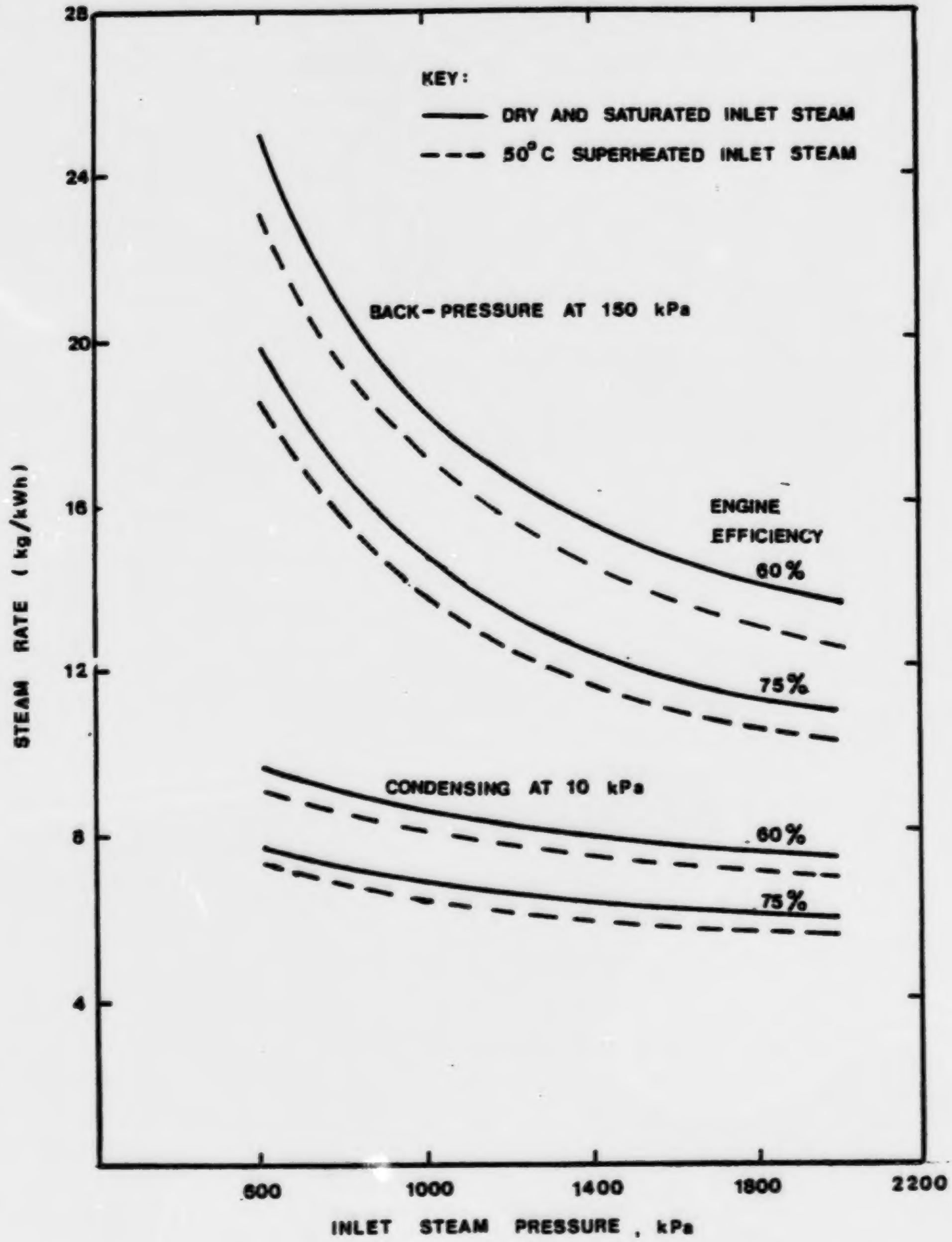


FIGURE VI-20 STEAM RATES FOR A STEAM ENGINE AT VARIOUS INLET AND EXHAUST CONDITIONS

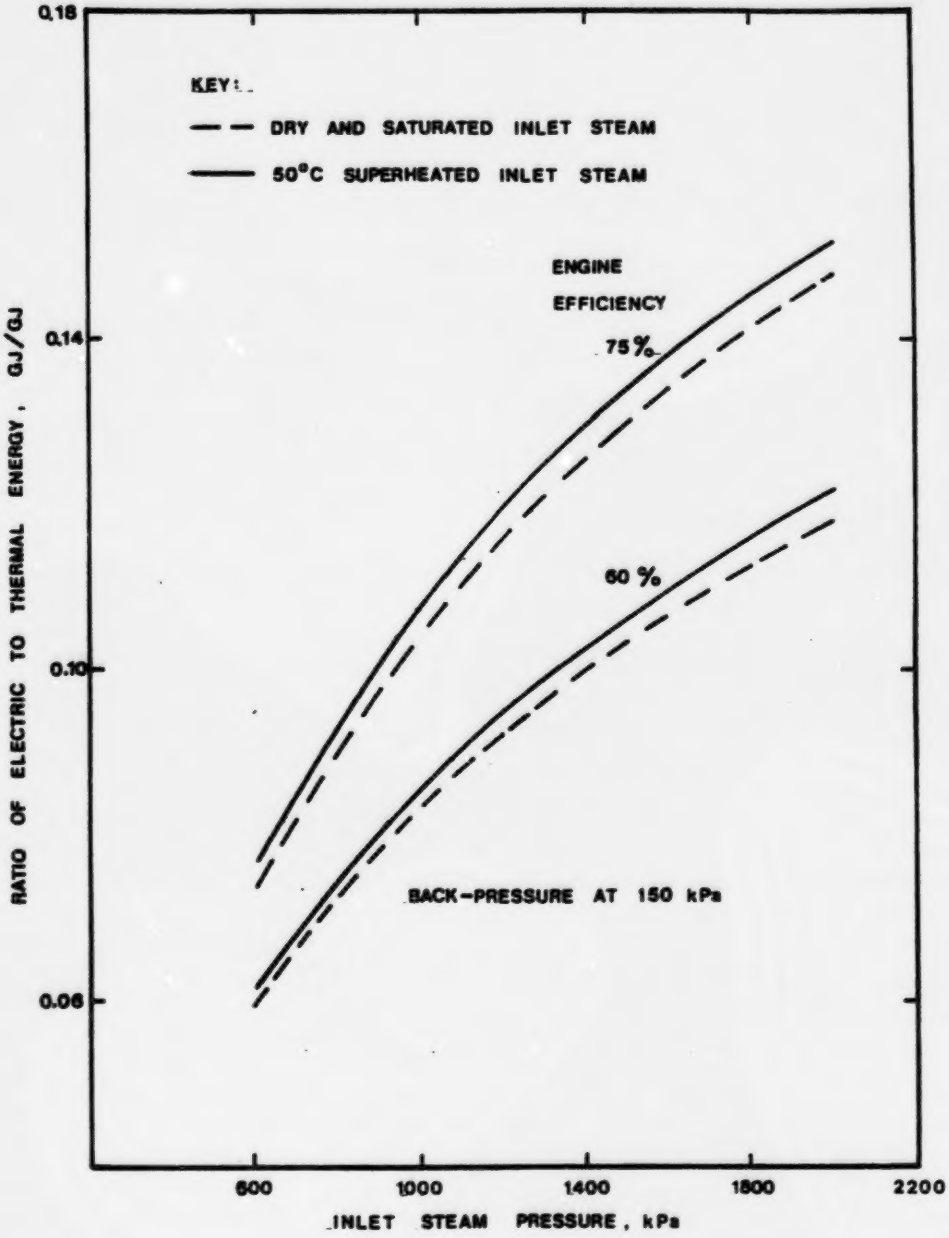


FIGURE VI-21 RATIO OF ELECTRICAL TO THERMAL ENERGY FOR STEAM ENGINE

Advantages:

- excellent torque characteristics and easily started under heavy loads;
- direction of rotation easily reversed;
- suitable for variable and slow speeds;
- in smaller sizes, more economical than the steam turbine.

Disadvantages:

- requires a large space per unit output;
- high thermal losses;
- requires a large flywheel to smooth out speed fluctuations during each revolution;
- contact between moving parts such as the piston and cylinder wall neccessitates internal lubrication;
- in larger sizes, not as economical as the steam turbine.

As far as economy, i.e. specific steam consumption, is concerned, a general rule to follow is that below 2 MW the steam motor is better; above 20 MW the steam turbine is preferred and between 2 MW and 20 MW a comparison study is required to determine which is best suited for the specific case.

Maintenance of steam engines is minimal at worst, sometimes having periods of ten years or more without major overhauls being performed. The engines are now constructed to allow easy access to components and readily changeable parts.

3. Manufacturers

a) Spilling-Werk A.G.

Spilling-Werk has been manufacturing steam engines for well over one hundred years. With the advent of the steam turbine, Spilling maintained their interest in developing the steam engine which has culminated in their concept of the modular steam engine. A modular engine motor is assembled by linking engine blocks together, each block consisting of a crankcase unit and a cylinder block. For example, by linking three engine block units together, one has a three cylinder steam motor.

There are basically two models of cylinder blocks each having the same overall dimensions and connection points so that they may interconnect regardless of model type. For each model there is a limitation as to the number of engine blocks that can be linked in one steam motor. Table VI-11 gives pertinent data on the two models of engine blocks.

The total output of a steam engine is simply the sum of the outputs of the individual cylinders. From the various combinations of the engine models, total outputs of between 25 and 2,000 kW can be obtained. In addition, after a steam engine has been placed in operation, its output can be increased by adding engine blocks in the field.

The engine can use steam at pressures of between 400 kPa and 15,000 kPa and exhausts steam generally between 150 kPa and 2,500 kPa. The steam motors can operate from single expansion to quadruple expansion, depending on steam conditions, thereby allowing steam to be extracted at various pressure for process use.

Cost is dependent on the number of cylinders, the output of the steam motor, or type of extraction. Generally, prices are based on specific cases, however, one example is that of a two cylinder back-pressure unit with an output of 400 kW costing approximately $250,000 ($625/kW), including auxiliary equipment such as polishers, governor and instrumentation.

b) Skinner Engine Company

Skinner manufactures traditional vertical steam engine-generator sets in sizes ranging from 5 kW to 3,000 kW output. Typically, these engines have engine efficiencies of approximately 65% and may be of single or multiple cylinder design.

The engine may be of condensing type or have back-pressures of up to 200 kPa. Typically, the engine operates with inlet steam at pressures of between 600 and 4,000 kPa, dry and saturated. The engines have low shaft speeds ranging between 320 and 600 rpm.

Typically, costs vary with size of engine and whether the engine is of back-pressure or condensing design. For example, a model E-9, duplex, 2-cylinder, vertical steam engine-generator set with an output of 100 kW would cost approximately $60,000 or $600 per kW, whereas a model DV-6 uniflow, six-cylinder, vertical steam engine-generator set costs approximately $1 million, or $500 per kW.

Table VI-11

PERFORMANCE DATA FOR SPILLING-WERK STEAM ENGINE

	ENGINE BLOCK MODEL	
	No. 7	No. 12
Generator Output per Cylinder (kW)	25	200
Maximum Units per Steam Motor	5	10
Maximum Engine Speed for 60 Hz Power (rpm)	1,200	900

F. Advanced Direct Conversion Systems

1. Fuel Cells

Fuel cells are electrochemical energy conversion devices which continuously transform the chemical energy of a fuel and oxidant to electrical energy. The process is carried out at constant temperature using an electrode-electrolyte system.

The complete fuel cell power-generating system consists of three components: the fuel processing system; the fuel cell; and the power conditioner. The fuel processor might be a reformer, a gasifier, a hydro-processor, or a combination of these, depending on the fuel used. In the case of wood, a gasifier is used to convert the solid fuel to a gas which contains hydrogen gas and carbon monoxide. The produced fuel gas must be cleaned to remove condensibles and sulphur, then upgraded to increase the hydrogen concentration.

The hydrogen-rich gas is then fed to the anode side of the fuel cell where it is ionized (Figure VI-22). The hydrogen ions react with oxygen from the cathode to form water and the electron flow from the anode to the cathode produces a voltage. The DC power produced is then transformed to AC power in a power conditioner.

There are several different types of fuel cells, but only two are close to being ready for commercial applications, the molten carbonate fuel cell and the phosphoric acid fuel cell. These fuel cells are so-named due to the electrolyte that is used. A major difference between the two is operating temperature, approximately 190°C to 210°C for the acid cell and 600°C to 700°C for the molten carbonate cell (Benjamin et al). In both cells the water produced from the reaction contains thermal energy which can be used for processing of the fuel, or for steam or heating requirements.

Fuel cells typically have flat electrical efficiency curves throughout the range of rated power levels, varying from 40 to 60% based on the lower heating value of the fuel. With cogeneration systems, overall efficiencies may range from 80 to 90% recovery. Table VI-12 shows typical efficiency values for the two types of cells of interest (Sperberg and Woods). Note that the molten carbonate fuel cell produces thermal energy at a much higher temperature than the phosphoric acid cell.

Current literature indicates that the phosphoric acid cell should be ready for commercial introduction by 1985 to 1986 (Sperbeg and Woods). On the other hand, the molten carbonate fuel cell will not have advanced to this stage until 1990 (Krumpelt and Ackerman).

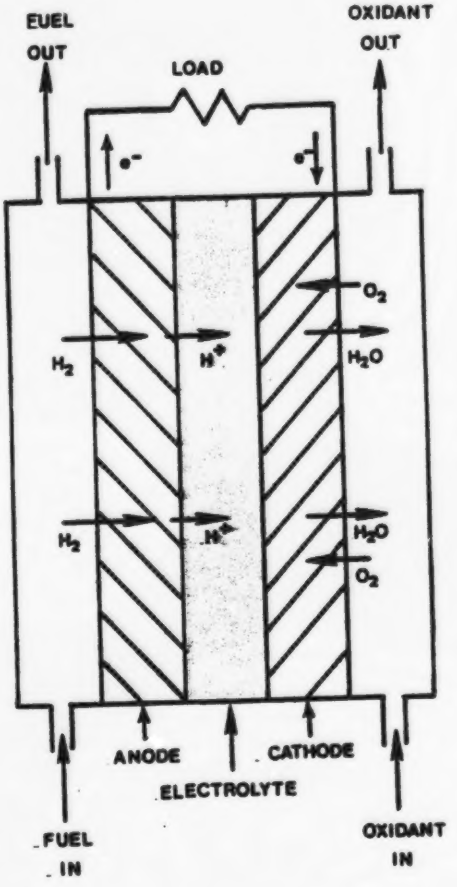


FIGURE VI-22 FUEL CELL CONFIGURATION

Table VI-12

PERFORMANCE DATA FOR FUEL CELLS

CHARACTER	PHOSPHORIC ACID	MOLTEN CARBONATE
Electrical Conversion Efficiency	40	40 - 60
Thermal Conversion Efficiency	40 - 45	45 - 30
Overall Energy Efficiency	80 - 85	85 - 90
Thermal Energy		
High Grade Temperature	135°C	371°C
Low Grade Temperature	82°C	121°C
High Grade %	20	80 - 100
Low Grade %	80	20 - 0
Capacity Range	20 - 1,000 kW	1,000 - 1,000,000 kW

The two major developers of fuel cell commercial technology and hardware are United Technologies Corp. (phosphoric acid fuel cell) and General Electric Co. (molten carbonate fuel cell).

2. Thermionic Converters

A thermionic converter is a direct-conversion device consisting of a hot emitter and a cooled collector separated by a low-loss plasma, usually ionized cesium vapour in a vacuum. In operation, the emitter is heated to a high temperature, usually greater than 1,500°C, and electrons are "boiled" off the emitter, cross the inter-electrode medium, and enter the collector with sufficient excess energy to deliver power to an external load (Figure VI-23). In addition to electrical power, the temperature of the collector is raised, and heat is rejected which can be used as a thermal-energy source. Typically, the collector temperature is between 650°C and 800°C.

Thermionic devices would generally be located in areas of high heat flux, usually on furnace walls or arranged around the periphery of a combustor. Heat rejected by the collector may be used to pre-heat the combustion air in an air heater. Fuel and the preheated air is fed to combustor, where ignition occurs and the emitter is raised to operating temperature. Combustion gases leave the combustor for use as thermal energy or for electricity generation.

It has been suggested that by using thermionic converters in conjunction with other topping cycles such as steam or gas turbines, increases of electrical power generation of up to 10% may be expected (Miskolvzy et al). Currently, thermionic converters operate at approximately 30% electrical efficiency, and since high quality thermal energy is rejected, overall efficiencies of 90% to 95% can be achieved (United Technologies Corp.).

Electrical to thermal energy ratios of thermionic arrays are expected to be in the order of 0.25. These systems are still in the early developmental stages and are not expected to be available for topping operations in the immediate future. Current cost estimates propose incremental capital costs for these devices of between $400 and $600 per kW for applications on very large central power plants.

Companies involved in the development of thermionic technology are Thermo Electric Corp., Brown Boveri Turbomachinery and Rasor Assoc. Inc.

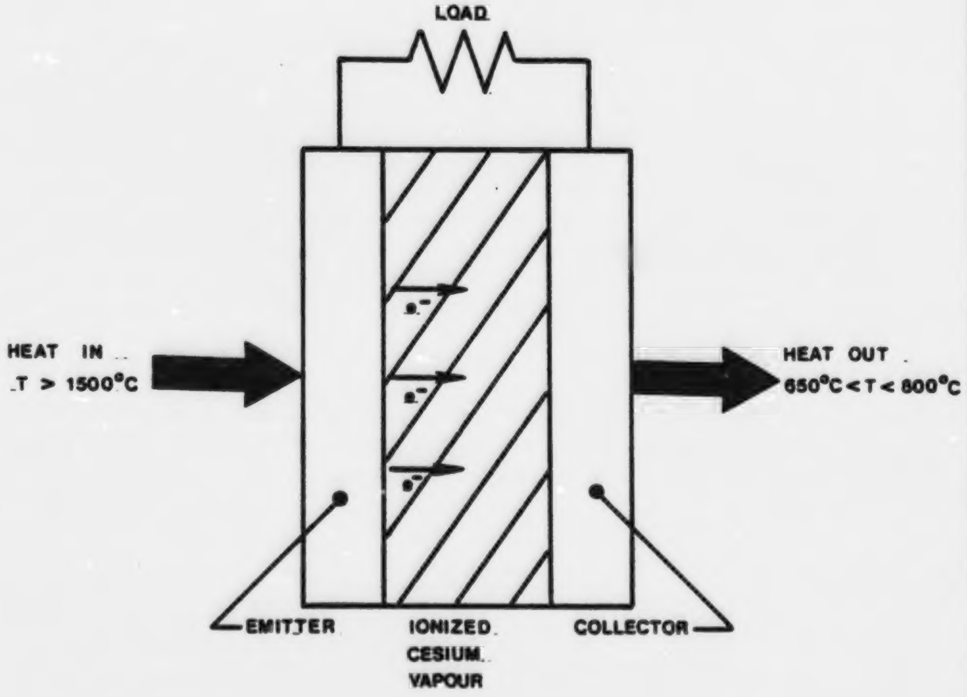


FIGURE VI-23 THERMIONIC CONVERTER

VII ELECTRICITY GENERATING SYSTEMS SUPPLIERS

A. Steam Boiler Systems

1. General

By judicious selection of a boiler design from the section covering wood conversion units, and a turbine-generator or steam engine-generator from the section covering prime movers, a preliminary design for any system of less than 5 MW output can be developed. This preliminary design process must make allowance for equipment range limitations, types of residues that are available, as well as site specific design factors such as equipment already in place. Boiler manpower requirements must be determined from provincial boiler acts.

In this section some of the information of steam driven electricity generating systems presented in detail elsewhere in this report is summarized. In addition, an indication is given as to the suppliers of complete steam boiler electricity generating systems.

2. Costs

Costs of steam driven electricity generating systems cannot readily be generalized because of the many variables affecting the selection of a boiler and turbine. A summary of costs is presented here to give budgetary figures adequate for comparison of alternative systems. Detailed cost comparisons of candidate systems is recommended prior to selection of a final design.

The traditional prime mover for electricity generation is a steam turbine. Steam engines may be used for this service, but few companies presently market them so availability and service may be poor.

Costs for wood fueled systems using either a steam turbine or a steam engine to generate 250 kW of electricity were estimated by SNC. Capital and operating costs for these systems are summarized in Table VII-1. These data indicate a total system capital cost of $4,544/kW with a steam engine, and $3,624/kW with a turbine. Operating costs for the two systems were very similar; 20.8¢/kWh for the steam engine; 19.2¢/kWh for the turbine. Both costs are above utility power costs. The major capital cost was the boiler which, at $1,308/kW, formed 36% of the turbine system cost. The costs for the turbine with generator ($324/kW) are above the values of $200 to 280/kW obtained in this study for a 500 kW unit with inlet steam at 4,000 kPa and 400°C because of the dependence of turbine cost on steam conditions.

Table VII-1

CAPITAL AND OPERATING COSTS FOR WOOD FIRED 250 kW STEAM TURBINE AND STEAM ENGINE SYSTEMS

	STEAM ENGINE	STEAM TURBINE
A. CAPITAL COSTS ($/kW)		
1. Boiler Plant 3-pass fire-tube boiler 4,500 kg/h, 1,551 kPa, 205°C Includes: fuel feed, combustion, water treatment, piping, tanks, controls, fans, stack	1,308	1,308
2. Prime Mover		
a) Two-cylinder Spilling-Werk Engine with generator (250 kW)	840	
b) Back-pressure turbine with generator		324
3. Fuel Handling for Chips	268	268
Subtotal	2,416	1,900
4. Building and Foundations	304	284
5. Miscellaneous Costs Includes: freight, insurance, delivery, assembly supervision	932	736
6. Engineering and Commissioning	288	228
7. Contingency	604	476
Total	4,544	3,624
B. OPERATING COSTS[a] (¢/kWh)		
1. Wages and salaries	6.8	6.8
2. Maintenance, insurance	0.8	0.6
3. Auxiliary power, lubricants	5.7	5.9
4. Financing - 15 years at 10%	7.5	5.9
Total	20.8	19.2

[a] Based on 8,000 h/a, or 2,000,000 kWh per year

The cost comparison of steam turbine and steam engine systems illustrates the advantage, based on capital cost, of the turbine systems. A 250 kW steam engine costs about $840/kW while a unit of about 400 kW costs about $625/kW. These costs are about 2.5 times the costs of comparably sized turbines.

Archibald and Gabriel have estimated the costs of back- pressure and condensing/extraction turbine systems with rated outputs of 2,500 kW and 5,000 kW, respectively. These results have been presented previously and will be only summarized here. They considered a cogeneration system with a boiler producing steam needed for processing. The base case is a 68,000 kg/h boiler generating steam at 1,720 kPa and 230°C. The incremental cost (Table VII-2) of upgrading the boiler to 4,130 kPa and 316°C and adding a back-pressure turbine of 2,500 kW rating is $915/kW. Because of the higher efficiency of a condensing turbine, the incremental cost of upgrading the system to generate 5,000 kW with a condensing/ extraction turbine is $703/kW. Because most of the boiler cost is assigned to process steam generation and not electricity generation, these incremental capital costs are considerably lower than for systems generating electricity only. Assume for example, a 30,000 kg/h boiler supplying all steam to a 5,000 kW condensing/ extraction turbine. At a cost of about $151/kg/h for the boiler, the system cost for boiler and turbine, complete, is $1,410/kW.

The costs presented by Archibald and Gabriel for the turbine and generator are in good agreement with data obtained during this work and with data updated from Noyes. The ratio of installed cost to turbine and generator cost ranges from 2.1 to 2.9 to one.

Costs for boilers are given in Chapter IV for most boiler sizes. Although the cost data was obtained from several sources, they appear self-consistent and are similar to values calculated by Archibald and Gabriel.

3. Manufacturers

Many boiler suppliers are capable of installing turnkey wood-fired electricity generating systems. These are not standardized, but require custom design to accommodate site requirements. Many boiler suppliers do not do turnkey projects, supplying the boiler under the supervision of an engineering consultant. Lists of boiler suppliers are available in several trade directories for the forest products industries.

During the selection of a steam boiler, both integral boilers (including combustion chamber) and combustor/boiler combinations should be considered. Most combustor manufacturers will supply systems for generation of steam up to about 2,000 kPa. This

Table VII-2

COSTS OF 2,500 kW AND 5,000 kW WOOD-FIRED SYSTEMS

	INSTALLED COST
A. BASE CASE:	
Boiler: 68,000 kg/h, 1,720 kPa, 230°C Includes: fuel handling, building, boiler, water treatment, scrubber, auxiliaries	$137/kg/h
B. INCREMENTAL COST FOR 2,500 kW BACK-PRESSURE TURBINE:	
1) Boiler upgrading to 4,130 kPa, 316°C	$ 14/kg/h $398/kW
2) Back-pressure turbine with generator 2,500 kW rated output Includes: building, turbine, generator, instrumentation, auxiliaries	$517/kW
Total	$915/kW
C. INCREMENTAL COST FOR 5,000 kW CONDENSING/EXTRACTION TURBINE	
1) Boiler upgrading to 4,130 kPa, 316°C	$ 14/kg/h $199/kW
2) Condensing/Extraction Turbine with generator 5,000 kW rated output Includes: building, turbine, generator, instrumentation, cooling tower, auxiliaries	$504/kW
Total	$703/kW

imposes restraints on the type of turbine that can be selected and its performance.

B. Gasifier and Pyrolyzer Systems

1. General

Although several systems for gasifiers were identified previously as alternatives, only those using the gas in an internal combustion engine or to generate steam in a boiler for expansion in a turbine or engine are commercially available. No internal combustion engine systems are presently well proven, although several are operating.

The evaluation of systems using an internal combustion engine coupled to a gasifier is presently hampered by the lack of test data for the contaminants in the fuel gas, particularly tar and particulate, and the lack of operating data for an engine operated on such a fuel. It is presently only possible to identify obvious potential problems such as tar deposition in intake manifolds and supply piping. Long-term effects of contaminants on lubricating oil, cylinder wear and carbon deposition on the cylinder head are not known.

Although several gasifier manufacturers are interested in or developing systems with internal combustion engines, few are presently marketing these systems. These will be identified in a following section on developers.

Generating steam for power generation using a wood-derived fuel gas can be considered as commercial technology. This system avoids gas cleanup and problems with evolved wood tars, but must satisfy boiler regulations and generally is suitable only when there is a need for process steam. Unless there is a need for fuel gas, or a gas-fired boiler is already in place, a combustor/boiler combination is recommended over a gasifier/boiler combination. One advantage of the use of a gasifier rather than a combustor is the ease of transporting the fuel gas around the plant.

None of the manufacturers that were contacted market gasifier/boiler combinations for power generation. Such a system would need to be custom designed.

Some very limited testing of firing a low heating value gas in a gas turbine has been done. Because of the lack of data for the content of contaminants in fuel gas from a wood gasifier, it is presently not possible to state with any certainty what problems might be encountered with a gasifier/gas turbine combination. Corrosion and fouling problems are likely unless extreme attention is payed to gas cleanup.

Considering the advantages and disadvantages of the various prime movers for gasifier systems, internal combustion engines are preferred. The engine may be of either single-fuel or dual-fuel design, the preference depending on the system reliability that is needed to satisfy mill operations or the utility, which ever is the most restrictive. If high reliability is essential, then a dual-fuel engine should be used.

Downdraft gasifiers are preferred for electricity generation systems of less than 1 MW output capacity. Larger systems could be assembled using a modular design. Although several problems have been encountered by operating systems, a downdraft gasifier is superior in this size range because of the low tar content of the fuel gas. This reduces gas cleanup difficulties, reduces problems with disposal of tar contaminated scrubber water and reduces tar deposition in the engine.

For systems above 1 MW, updraft gasifiers could be used, possibly up to output capacities over 2 MW, but the high tar content of the fuel gas and the low grate energy release rate makes them relatively unattractive compared to fluid-bed gasifiers. One updraft gasifier system of 1.8 MW capacity is operating successfully.

For systems over 1 MW, fluid-bed gasifiers are preferred. Considerable work is needed to develop a system using a fluid-bed gasifier, primarily with regard to gas cleanup and system control. The limited data that is presently available indicates that less tar will need to be removed from gas from a fluid-bed unit than from an updraft unit.

Gasifier systems could be completely automated and, thus, require less supervision than steam systems. Dual-fuel engines are very attractive fo systems where a backup fuel is needed to ensure continued power generation.

2. Costs

Equipment costs for most of the systems identified for generating electricity with a wood gasifier can be estimated from data present in the discussion of unit processes. Data obtained from gasifier manufacturers for complete systems are presented here to augment information presented in the previous sections.

Costs for systems depend on system size. Generally, units having output capacities of several hundred kilowatts cost $1,200 to $1,500/kW with gasifier, gas cleanup and engine-generator set included. Capital costs decrease from this level to about $800 to $1,000/kW as system output capacities reach 5 to 6 MW. Costs vary with type of gasifier due to inherent limitations on the maximum size of a single gasifier and to the sophistication of the gas cleanup system needed to produce a suitably clean gas.

Capital costs for systems can readily be determined from the data for gasifiers and internal combustion engines. For example, consider a 1 MW installation using a dual-fuel engine and a downdraft gasifier. The engine cost for diesel-fuel operations with water cooling is about $220/kW. De-rating by 25% for operation on fuel gas and allowing 30% excess capacity increases this cost to $420/kW. Typically, this engine would run at part load with an efficiency of 25% and a fuel gas supply of 18 GJ/h. Using the gasifier cost for a Biomass Corp. downdraft gasifier ($25,000 per GJ/h) the gasifier adds $450/kW to the capital cost, bringing the total to $870/kW. Allowance must be made for additional gas cleanup and engine related costs beyond what is included in the cost factors, as well as for miscellaneous system components. Assuming these costs to be 20% of the cost of primary equipment bring the total to $1,045/kW, a value comparable to manufacturers' figures.

3. Manufacturers and Developers

Table VII-3 summarizes the systems under development for generation of electricity from wood residues. All use internal combustion engines as prime movers, although some have investigated gas turbines. None of the companies market standard commercially proven systems, most being in stages of development ranging from conceptual designs to demonstration units. Systems indicated as demonstration/commercial in status are those which have been either installed at a business, but are being improved, or demonstrated at a test facility and are awaiting buyers.

The most common difficulties encountered with the gasifier/diesel generating systems are associated with automated feeding, gas cleanup and engine speed control.

C. Non-Pressurized Combustor Systems

1. General

Technology exists for coupling a combustor to either an indirectly fired gas turbine, a Stirling engine or a boiler subsequently connected to a steam engine or turbine. Of these alternative, only the steam-driven generating system could be assembled with commercial equipment. The combination of a combustor and a heat recovery boiler is an alternative to the use of an integral steam boiler discussed previously. Such a design gives considerable flexibility and is considered practical for systems generating less than about 50,000 kg/h steam.

Table VII-3

SUMMARY OF GASIFIER/INTERNAL COMBUSTION ENGINE SYSTEMS

MANUFACTURER			INSTALLATIONS AND REMARKS
Biomass Corp.			-110 kW unit in Anchorage, Alaska operated by Alaska Electric Cooperative. Problems with gas cleanup, ash carryover. Problems being resolved. -110 kW unit in Zim, Minnesota at Iron Ranch Resources Rehabilitation -Test units of 120 and 130 kW operated at head office. Ash carryover and tar-plugging of cleanup system resolved. 300 h engine test showed no engine wear. Tar disposal problem. Need improved engine air/fuel control.
	GASIFIER TYPE	downdraft	
	POWER OUTPUT (kW)	100 - 150	
	STATUS	demonstration/commercial	
	CAPITAL COST ($/kW)	(U.S.)1,000 - 1,600	
D.M. International			-Company designs systems to own specifications. -No units operating.
	GASIFIER TYPE	updraft	
	POWER OUTPUT (kW)	up to 2,100	
	STATUS	preliminary design	
	CAPITAL COST ($/kW)	(U.S.)2,400 (installed)	
Duvant			-Developed new GA-20 design. Demonstrated at head office. 250 h operation successful. Dual-fuel operation only using Duvant engine. Lockhopper feeding system. Wet scrubber and impact filter used for gas cleanup. Four units reportedly sold to Philippines.
	GASIFIER TYPE	downdraft	
	POWER OUTPUT (kW)	480 and 720	
	STATUS	demonstration	
	CAPITAL COST ($/kW)	(Can.)1,000 - 1,500	
ERCO			-Tests planned at head office using pilot gasifier and diesel engine. Details not available.
	GASIFIER TYPE	fluid-bed	
	POWER OUTPUT (kW)	225 - 350	
	STATUS	pilot	
	CAPITAL COST ($/kW)	N/A	
EZ Manufacturing			-250 kW unit operating at head office. Fired with wood chips and sawdust. Packaged system uses wet scrubber for gas cleanup.
	GASIFIER TYPE	stirred-bed	
	POWER OUTPUT (kW)	250	
	STATUS	demonstration/commercial	
	CAPITAL COST ($/kW)	N/A	

Table VII-3 (continued)

SUMMARY OF GASIFIER/INTERNAL COMBUSTION ENGINE SYSTEMS

MANUFACTURER		INSTALLATIONS AND REMARKS
Halycon Assoc.		-2,000 kW unit operating at United Timber, North Anson, Maine. Operating successfully at 1,600 kW. Gasifier modified by United Timber. Nordberg engines. Secondary gas cleanup by Cooper Energy of New Orleans, LA. Fuel is unhogged bark. Electricity sold to utility.
GASIFIER TYPE	updraft	
POWER OUTPUT (kW)	1,500 - 2,000	
STATUS	demonstration	
CAPITAL COST ($/kW)	N/A	
Imbert		-50 kW portable unit tested by Saskatchewan Power Corp. Problems with feed, gas cleanup, ash removal on original unit. Deutz air-cooled engine functioned well and showed no wear at end of test. Feed was by batch loading using chips. Some problems with solid flow in gasifier. -120 kW unit operating at Snowshoe Industries, Fort Providence, N.W.T. Problems with automated feed system and engine speed control. Single fuel water-cooled MWM engine used. Gasifier firing pine wood chips. Plan continuous automatic operation.
GASIFIER TYPE	downdraft	
POWER OUTPUT (kW)	50 - 500	
STATUS	demonstration/commercial	
CAPITAL COST ($/kW)	120 kW - 1,250	
Omnifuel		-Tests planned at Levesque Plywood, Hearst, Ontario, using existing gasifier and 500 kW engine. Gas cleanup system to be developed. Have studied systems of up to 6,000 kW.
GASIFIER TYPE	fluid-bed	
POWER OUTPUT (kW)	500 - 6,000	
STATUS	pilot	
CAPITAL COST ($/kW)	6,000 kW - 800	
Pillard		-Systems developed in France and installed outside North America Available with automatic feed. Wood chips and charcoal can be fired. Common problems have been feed automation, automatic control and gas cleanup.
GASIFIER TYPE	downdraft	
POWER OUTPUT (kW)	20 - 500	
STATUS	demonstration/commercial	
CAPITAL COST ($/kW)	50 kW - $1,848 500 kW - $1,520*	

Table VII-3 (continued)

SUMMARY OF GASIFIER/INTERNAL COMBUSTION ENGINE SYSTEMS

MANUFACTURER		INSTALLATIONS AND REMARKS
Pyrenco		-120 kW unit operated for 500 h. Gasifier fired with wood pellets. Caterpillar dual-fuel turbocharged and naturally aspirated engines used. No wear of engine observed.
GASIFIER TYPE	downdraft	
POWER OUTPUT (kW)	120 - 1,500	
STATUS	demonstration	
CAPITAL COST ($/kW)	N/A	
Standard Solid Fuels		-Investigating gas turbines, engines and steam cycles for generation of electricity using SSF gasifier. Propose 2,000 kW system with wet scrubber for gas cleanup and compessor prior to delivery of gas to diesel engine.
GASIFIER TYPE	stirred-bed	
POWER OUTPUT (kW)	2,000	
STATUS	planned prototype	
CAPITAL COST ($/kW)	N/A	

*Includes 250 kW engine standby for operation on diesel fuel.

Most of the discussion presented under the heading of Steam Boiler Systems also applies to combustor/boiler combinations.

Use of Stirling engines with wood combustors is hampered by the lack of any engine in sizes above 40 kW. Use of a Stirling engine is attractive because there is no contact between moving engine parts and the hot combustion gases.

The type of combustor best suited to a given fuel and operating requirement can be determined from data presented in the discussion of combustors included in the Chapter on wood conversion process units. For dry, fine fuel the most attractive combustor is a suspension burner. These are available in a wide range of sizes and have a good record of operation. For wet, coarse fuel either a multiple chamber or fluid-bed combustor may be used, both of which have similar output capacities and fuel quality restrictions. Fluid-bed units have higher power requirements and slightly higher capital costs than many of the multiple chamber units. Particulate emissions from the combustor are highest for the fluid-bed designs.

2. Manufacturers

Many manufacturers of multiple chamber and fluid-bed combustors will supply the combustor complete with a boiler. Boilers having steam pressures up to 2,000 kPa are readily available. Higher pressure boilers generally must be special orders.

D. Pressurized Combustor Systems

Pressurized external combustors may be used to fire gas turbines directly. These systems have been more for coal-fired combustors than wood-fired combustors. Research and development work has been conducted in Britain, the United States and Australia to investigate the effects of coal ash particulates entrained in the combustion gases on the endurance of turbine blades and stator.

Two companies were identified which have experience with firing a gas turbine directly with gases from a pressurized combustor. These are Combustion Power Co. and Aerospace Research Corp.

1. Combustion Power Co.

A development program was undertaken by Combustion Power Co. under the sponsorship of the Environmental Protection Agency, Office of Research and Development (contracts 68-03-0054 and -0143). The main objective objective of this work was the development of a 1 MW system fueled by solid-waste in which a fluid-bed combustor supplied energy to drive a direct-fired open-cycle gas turbine. The combustor was pressurized to a level required by the turbine. This program led to development of an electrostatic granular filter which has since been patented and is available commercially from the company.

Limited testing was done with the combustor fueled by wood residues (Moody). During the tests, severe problems were encountered with deposits of particulate in the turbine. To complete work at that time, this problem was solved by periodic injection of milled walnut shells.

The company has proposed a wood-fired system similar to the one tested in the CPU-400 program, but incorporating an electrostatic granular filter as the second stage of gas cleanup. The combustor would be pressurized so that the hot combustion gases could be expanded directly through a gas turbine. In 1976, the costs of a system including a fluid-bed combustor, gas cleanup equipment and a gas turbine were estimated to be $500/kW. No tests of a system of this type have been conducted since the work for the EPA was completed in 1968.

2. Aerospace Research Corp.

This company is conducting research and development work funded by the U.S. Department of Energy to summarize the findings of past experimental trials of gas turbines fired with pressurized coal combustors and to conduct tests of gas turbines fired similarly with a wood combustor (Hamrick). The accomplishments to date have been analysis of past work with coal-fired turbines, engineering design and economic analysis of two combined cycle gas turbines, estimates of wood supplies for gas turbine power generation and a test of the gas turbine and wood combustor.

The test facility consists of a 500 hp gas turbine coupled to a compressor which supplies air to a specially designed external wood combustor. The combustion gases are passed through cyclone filters prior to expansion through the turbine. The test program will involve operation for 100 to 200 hours of a turbine with trial blade and stator materials, followed by dis-assembly and inspection of the turbine for corrosion and erosion. The combustor will be fueled by pulverized yellow pine wood.

Work under this contract is continuing.

E. Bottoming Heat Recovery Boiler Systems

An extensive list of the manufacturers and developers of bottoming organic Rankine systems has been presented by Curran in a 1980 publication. From this complete list, the ones used for generating electricity were extracted, producing the information in Table VII-4.

The prime movers in these systems are predominantly turbines. The fluid related problems reported by the manufacturers vary from no observed problems to corrosion and fluid decomposition. With proper selection of the fluid, most corrosion and decomposition problems can be avoided.

Table VII-4

PARTIAL LIST OF MANUFACTURERS OF BOTTOMING ORGANIC RANKINE SYSTEMS FOR POWER GENERATION

MANUFACTURER	POWER (kW)	HEAT SOURCE	TOTAL OPERATING TIME (h)	LOCATION
MTI	335	Steam generated by Diesel exhaust	N/A	Rockville Centre, NY
Nigata Iron Works	60	Steam generated by Diesel exhaust	N/A	Toyko, Japan
Sundstrand	600	Waste gas	N/A	Various
Mitsui	670	Furnace exhaust	N/A	Fukuyama, Japan
Thermo-Electron	450	---	>100	Waltham, MA
Institute di Macchine	40	Exhaust gas	300	Milan, Italy
Ormat Turbines	1 - 6	Gas and oil	18×10^6	Worldwide

Source: Curran

VIII STATUS OF SYSTEMS FOR ELECTRICITY GENERATION

A. Commercial Systems

In total, nineteen alternative configurations for electricity generation from wood were considered. Of these, only the systems using steam-driven turbines or engines can be considered as proven commercial technology. Steam engines are included in this assessment, even though very limited experience with the engines has been gained in recent years, and only two suppliers of the engines were identified. It has been assumed that with documented past experience with steam engines and the abundance of designers experienced with steam turbines, a steam engine generator set could be readily installed in Canada.

Of the remaining systems not using steam as the working fluid, the one nearest to commercial status is the combination of a wood gasifier and Diesel engine. Most manufacturers are presently either developing or demonstrating this technology. With market demand, proven standard systems with outputs of up to 3 MW and possibly up to 5 MW could be available within the next one or two years.

Weak points of existing systems are the feed system, the gas cleanup equipment and the system for regulation of the engine fuel supply rate. Automated feed systems have suffered from control problems and poor reliability. Gas cleanup equipment has functioned adequately in most cases, but improvements are needed. Technology for improving the design of gas cleanup equipment could be borrowed from other industries treating similar products, such as industries processing coal-derived gas and chemical products. Improvement of engine speed control requires attention to the gasifier design to achieve steady gas quality, and allowance for variability of the gas composition and heating value by incorporation of a mixing chamber to reduce the effects of transient quality flucuations.

B. Emerging Systems

Several systems have potential, but need further development. One of these is a combination of a gasifier with a direct-fired gas turbine. Presently, this system appears attractive for systems larger than suitable for diesel engines, possibly in excess of 4 to 5 MW. No turbines are available that can guarantee performance on wood-derived fuel gas. Costs of these units, therefore, cannot be estimated with any confidence, but appear to be higher than diesel engines.

Indirectly-fired gas turbines recovering heat from fluid-bed or other combustors is nearing the demonstration stage. Two possibilities exist, one being heat removal from within the bed of a fluid-bed combustor using a metal heat exchanger, the other being heat removal from the combustion gases using either a metal or ceramic heat exchanger. The ratio of electric to thermal energy of these systems makes these systems unattractive for all forest products industries but pulp and paper.

The more attractive of the emerging systems is one coupling a combustor or gasifier to a Stirling engine. Assessment of this technology in detail is recommended. Because of the limited output capacity of available engines, the most attractive installations at present are ones having outputs of less than about 300 kW.

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APPENDIX A - LIST OF COMPANIES

FIRM	PROCESS EQUIPMENT
Abco Industries, Inc. 2675 E. Hwy. 80 Abilene, TX U.S.A.	Steam Boilers
Advanced Combustion Inc. P.O. Box 36 Don Mills, Ontario M3C 3G5	Steam Boilers
AEG-Kanis Turbinfabrik GMBH Postache 1040 8500 Nurnberg 1 West Germany Phone (0911) 4130-1 Telex 06-22731	Steam and Gas Turbines
Alberta Industrial Developments Ltd. 704 Cambridge Building Edmonton, Alberta T5J 1R9	Fluid-Bed Gasifiers and Pyrolyzers
Alturdyne 8050 Armour Street San Diego, CA. 92111 U.S.A.	Gas Turbines
Ange Varma Canada Ltd. 1081 Barton Street Thunder Bay, Ontario P7B 5N3	Multiple Chamber Combustors
APE Canada Inc. 2333 Millrace Ct. No. 4 Mississauga, Ontario L5N 1W2	Steam Turbines
Applied Engineering 1525 Charleston Hwy. Orangeburg, SC 29115 U.S.A.	Updraft Gasifiers

Babcock & Wilcox Canada Ltd. 3031 Viking Way Richmond, B.C. V6V 1W1	Steam Boilers
Basic Environmental Engineering 21 W161 Hill Avenue Glen Ellyn, IL 60137 U.S.A.	Multiple Chamber Combustors
Berg & Stark A.B. Brunette Machine Works Ltd. No. 5 Capilano Way New Westminster, B.C. V3L 5G3	Multiple Chamber Combustors
Bigelow-Liptak of Canada Ltd. 1831 Mattawa Avenue Mississauga, Ontario L4X 1K7	Single Chamber Combustors
Biomass Corp. 951 Live Oak Blvd. Yuba, CA 95991 U.S.A.	Downdraft Gasifiers
Brown Boveri Turbomachinery, Inc. 711 Anderson Avenue North St. Cloud, MN 56301 U.S.A.	Gas Turbines
Bumstead - Woolford Co. P.O. Box 448 Woodinville, Washington 98072 U.S.A.	Suspension Burners
Caterpillar Marketing P.O. Box 3900 Peoria, IL 61614 U.S.A.	Gas and Diesel Engines
Century Research, Inc. 16935 S. Vermont Avenue Gardena, CA 90247 U.S.A.	Updraft Gasifiers
Cleaver Brooks of Canada Ltd. 161 Lorne Avenue Stratford, Ontario N5A 6T8	Steam Boilers

CNB Tri-Fuel Boiler Division of Combustion Service & Equipment Co. 2016 Babcock Boulevard Pittsburg, PA 15209 U.S.A.	Steam Boilers
Coen Company 1510 Rollins Road Burlingame, CA 94010 U.S.A.	Suspension Burners
Combustion Power Company, Inc. 1346 Willow Road Menlo Park, CA 94025 U.S.A.	Fluid-Bed Combustors
Consumat Systems Inc. P.O. Box 9579 Richmond, VA 23228 U.S.A.	Single Chamber Combustors
Coppus Engineering Corporation 344 Park Avenue Worcester, Mass. 01610	Steam Turbines
Cummins Rio Grande Sales & Service Ltd. 1921 Broadway N.E. Albuquerque, NM 87102 U.S.A.	Gas and Diesel Engines
C. E. Canada 99 Bank Street, Ste. 501 Ottawa, Ontario K1P 6C5	Steam Boilers
Deltak Corporation P.O. Box 9496T Minneapolis, MN 55440 U.S.A.,	Steam Boilers
Detroit Diesel Allison Ltd. P.O. Box 894 Indianapolis, IN 46206 U.S.A.	Gas Turbines
Deutz Diesel (Canada) Ltd. 90 Montee DeLiesse St. Laurent, Quebec H4T 1N4	Gas and Diesel Engines

D.M. International Inc. 6161 Savoy Drive P.O. Box 36444 Houston, TX 77036 U.S.A.	Updraft Gasifiers
Eclipse Fuel Engineering Co. of Canada Ltd. 20 Upjohn Road Don Mills, Ontario M3B 2V9	Steam Boilers
Econo-Therm Energy Systems Corp. 11535A K-Tell Drive Minnetonka, MN U.S.A.	Steam Boilers
Elliott Steam Turbines 1111 Queen Street West Brampton, Ontario V6Y 1M3	Steam Turbines
Energy Products of Idaho 3805-T Industrial Avenue S. Coeur D'Alene, ID U.S.A.	Fluid-Bed Combustors, Steam Boilers, Fluid-Bed Gasifiers
Energy Resources Company Inc. 185 Alewife Brook Parkway Cambridge, MA 02138 U.S.A.	Fluid-Bed Gasifiers and Pyrolyzers
EZ Manufacturing Company RD1 New Holland, PA 17557	Stirred-Bed Gasifiers
FluiDyne Engineering Corp. 5900 Olson Memorial Highway Minneapolis, MN 55422 U.S.A.	Fluid-Bed Combustors and Heat Exchangers
Foster Wheeler Energy Corp. 110 South Orange Avenue Livingston, NJ 07039 U.S.A.	Energy Conversion Systems
Foster Wheeler Ltd. 81 Eastchester Avenue P.O. Box 3007 ST. Catherines, Ontario L2R 7B7	

Garrett Corporation Industrial Engines and Products Box 5217 Phoenix, AX 85010 U.S.A.	Gas Turbines
General Electric GT Div. One River Road Schenectady, NY 12345	Gas Turbines
Georgia-Pacific Corporation 7920 S.W. Hunziker Road Tigard, OR 97223 U.S.A.	Suspension Burners
Guaranty Performance Co. Inc. P.O. Box 230045 Tigard, OR 97223 U.S.A.	Suspension Burners
Hague International 3 Adams Street South Portland, Maine 04106 U.S.A.	Ceramic Heat Exchangers
Halcyon Associates, Inc. Maple Street East Andover, NH 03231 U.S.A.	Updraft Gasifiers
Herman Bogot & Co. 3143 N. Nottingham Avenue Chicago, IL 60634 U.S.A.	Steam Boilers
Industrial Boiler Company, Inc. P.O. Box 2258 Thomasville, GA 31792 U.S.A.	Steam Boilers
Industrial Boiler Co. P.O. Box 936 221 Law Street Thomasville, GA 31792 U.S.A.	Steam Boilers
Johnston Boiler Company Ferrysburg, MI 49409 U.S.A.	Steam Boilers

Kerr Combustion Int'l Inc. P.O. Box 386 Oakville, Ontario L6J 5A8	Steam Boilers
Kongsberg Vapenfabrik Gas Turbine Division P.O. Box 25 N-3601 Kongsberg, Norway	Gas Turbines
Konus Systems Inc. Western Canada QM Energy Systems P.O. Box 1868 Prince George, B.C. V2L 5L3 Eastern Canada WM. Edgett & Sons P.O. Box 202 Orillia, Ontario	Multiple Chamber Combustors
Lamb-Cargate Industries Ltd. P.O. Box 440 New Westminster, B.C. V3L 4Y7	Multiple Chamber Combustors
Lockhead-Haggerty Engineering & Mfg. Co. Ltd. 1027 Royal Avenue New Westminster B.C. V3M 1K3	Suspension Burners
McConnell Industries P.O. Box 26210 Birmingham, Alabama U.S.A.	Suspension Burners
Mellinger, E.R. 108 Carmathon Street Saint, John, N.B.	Updraft Gasifiers
Mottpac Boilers Inc. 1028 Royal York Road Toronto, Ontario MBX 2G4	Steam Boilers
MWM Murphy Stevenson 5317 West Burnham Street Milwaukee, WI 53219 U.S.A.	Gas and Diesel Engines

Nebraska Boiler Company, Inc. 70th & Cornhusker Hwy. Lincoln, NE U.S.A.	Steam Boilers
Olivine Corporation 1015 Hilton Avenue Bellingham, WA 98225 U.S.A.	Single Chamber Combustors
Omnifuel Gasification Systems Ltd. 3284 Yonge Street Ste. 501 Toronto, Ontario M4N 3M7	Fluid-Bed Gasifiers
Onan Int'l Power Products Division 1400 73rd Avenue N.E. Minneapolis, MN 55432 U.S.A.	Gas Turbines
Peabody Gordon Piatt P.O. Box 650 Winfield, Kansas 67156 U.S.A.	Suspension Burners
Pendell Boiler Ltd. 50 Bartlett Avenue Toronto, Ontario M6H 3E6	Steam Boilers
Pillard Inc. 8001 Franklin Drive Farms Suite 207, Koger Building Richmond, VA 23288 U.S.A.	Downdraft Gasifiers , Suspension Burners, Packaged Power Generating Plants for Wood
Pratt & Whitney Aircraft of Canada Ltd. Box 10 Longueuil, Quebec J4K 4X9	Gas Turbines
Ray Burner Company 1301 San Jose Avenue San Francisco, CA 94112 U.S.A.	Steam Boilers and Energy Generation Systems

Ruston Diesels Ltd. Vulcan Works Newton-Le-Willows Merseyside, England WA12 8RU	Gas and Diesel Engines
SACM Diesel 1, Rue de la Fonderie BP 1210 F 68054 Mulhouse Cedex Phone (89) 46.01.08 Telex 881699F	Gas and Diesel Engines
U.S. Agent F.W. Donnelly Co. 1939 West Gray Houston, Tx 77019	
Canadian Agent Western Diesel Sales & Mfg Ltd. 25 Victoria Drive Vancouver, B.C. V5L 4C1	
Saskatchewan Power Corp. 2025 Victoria Avenue Regina, Sask. S4P 0S1	User of Imbert and Fluid-Bed Gasifiers
Saskatoon Boiler Mfg. Co. Ltd. 2011 Quebec Avenue Saskatoon, Sask. S7K 1W5	Steam Boilers
Skinner Engine Company P.O. Box 1149 Erie, PA 16512 U.S.A.	Steam Engines
Spilling-Werk Wertstrasse 5, 2 Hamburg 11, West Germany Phone 040/781335 Telex 2163419	Steam Engines
Standard Solid Fuels, Inc. P.O. Box 1389 Everett, WA 98206 U.S.A.	Stirred-Bed Gasifiers

Stirling Power Systems 7101 Jackson Road Ann Arbor, Micigan U.S.A.	Stirling Engines
Synchro Combustion Equipment Ltd. P.O. Box 669 Seaforth, Ontario	Steam Boilers
T R Systems Inc. 237 Commerce Street Glastonbury, CT 06033 U.S.A.	Combustors
Terry Steam Turbine Company Division of Terry Corp. Lamberton Road Windsor, CT 06095 U.S.A.	Steam Turbines
The Bethlehem Corp. 25th & Lennox Streets P.O. Box 348 Easton, PA 18042 U.S.A.	Steam Boilers
The Gaskell Company, Inc. P.O. Box 13225 Memphis, TN 38113 U.S.A.	Steam Boilers
The McBurney Corporation P.O. Drawer 47848 Atlanta, GA 30362 U.S.A.	Steam Boilers
The O'Brien Machinery Co. Green and Washington Streets Dowingtown, PA 19335 U.S.A.	Gas and Diesel Engines and Indirectly Fired Gas Turbines
Thermo Electron Corp. Energy Systems 101 First Avenue Division T Waltham, Mass. 02154 U.S.A.	

Ultrasystems, Inc. 2400 Michelson Drive Irvine, CA 92664 U.S.A.	Steam Boilers
United Stirling Inc. 11169 Tattersall Trail Oakton, Va 22124 U.S.A. Attention: Mr. W. Percival	Stirling Engines
Volcano Inc. 2020 Ste. Anne St. Hyacinthe, Quebec J2S 5H2	Steam Boilers
Vyncke Canada Ltd. P.O. Box 270 Cremona, Alberta	Steam Boilers
Waycot Systems Ltd. 7 King Street East Suite 1604 Toronto, Ontario M5C 1A2	Suspension Burners
Wellons Inc. P.O. Box 381 Sherwood, OR 97140 U.S.A.	Steam Boilers
Westinghouse Canada Ltd. 286 Sanford Avenue N. Hamilton, Ontario	Steam and Gas Turbines
Westwood Polygas Ltd. 1444 Alberni Street Vancouver, B.C. V6G 1A1	Updraft Gasifiers
White Engines, Inc. 101 Eleventh Street, S.E. Canton, OH 44707 U.S.A.	Gas and Diesel Engines
Witermo Oy Marjatta Aalto Inc. 150 - 500 Duplex Avenue Toronto, Ontario M4R 1V6	Multiple Chamber Combustors

Worthington (Canada) Ltd. 320 - 145 West 17th Street North Vancouver, B.C. V7M 1V5	(Turbodyne Steam Turbines)
York-Shipley of Canada 1297 Industrial Road Cambridge, Ontario N3H 4S6	Steam Boilers